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THESIS

EVALUATION OF SYSTEM ARCHITECTURES FOR THE ARMY AVIATION GROUND POWER UNIT

by

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December 2014

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EVALUATION OF SYSTEM ARCHITECTURES FOR THE ARMY AVIATION GROUND POWER UNIT

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Ground support equipment is critical to the success of Army Aviation. As the Aviation Ground Power Unit evolves or is replaced, it will be necessary to reduce life cycle costs and improve availability. This thesis explores the requirements and offers potential architectures and component selection to satisfy the Army Aviation Ground Power Unit requirements while increasing value. Using the current system as a baseline, alternatives were compared using performance, mass, envelope, reliability, and life cycle costs. The power plant proved to be the most important component in the architectures examined. Power plant influence on the life cycle cost of the system was the dominant factor among the selection criteria; fuel and power plant maintenance costs were the largest contributors to system life cycle costs. The research concludes that architectures with diesel engine power plants are preferred even though these architectures have an inherent mass risk and require greater interaction between aviation and ground maintenance activities.

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EXECUTIVE SUMMARY

The Army's Aviation Ground Power Unit (AGPU) was placed into service in 1984 to provide electrical, hydraulic, and pneumatic power for the Army rotary aircraft fleet. As the primary piece of aircraft support equipment, the AGPU is critical to the effectiveness of Army aviation in the tactical environment. This research examines architectures and component selection to provide required functionality at reduced life cycle cost and better to integrate the AGPU into the maintenance support infrastructure.

The results of the study showed that specific fuel consumption of the power plant is the predominant factor in scoring of the alternatives followed by the procurement cost of the power plant. The dominant architecture proves to be a diesel power plant substituted into the current AGPU design. Although this architecture has only 20% margin against the maximum mass requirement, the system exhibits a significant reduction in life cycle cost against the other alternatives.

The preferred alternative is estimated to have a minimum life cycle cost reduction of $170,700 per unit in CY2014 US dollars at a confidence level of 80%.

Mass, envelope, and reliability estimates for three alternate architectures are developed based on the performance requirements and compared against the current architecture. Procurement as well as operation and support costs are estimated for each system and used to generate a net present value life cycle cost point estimate. The net present value analysis is performed using a hypothetical program of 720 units, procured and placed into service over a five-year period with an assumed product service life of 20 years. A sensitivity analysis of the cost projections is performed using the Enhanced Scenario-Based Method. Higher cost variances are used for the diesel architectures to accommodate potential expenditures related to system mass risk mitigation.

An evaluation of the alternatives using weighted parameters consisting of performance, mass, envelope, noise, reliability, and life cycle cost is being performed to establish the favored alternative. Systems with diesel power plants rate higher than systems with gas turbine power plants. The baseline system is the least preferred.

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LIST OF ACRONYMS AND ABBREVIATIONS

AGPU	Aviation Ground Power Unit
AGSE	Aviation Ground Support Equipment
API	American Petroleum Institute
APU	auxiliary power unit
ATP	Acceptance Test Procedure
bhp	brake horsepower
CER	cost estimating relationship
COTS	commercial off-the-shelf
CV	coefficient of variation
DDH	direct drive hydraulics
DLA	Defense Logistics Agency
DMWR	Depot Maintenance Work Requirement
EDH	electric drive hydraulics
eSBM	Enhanced Scenario-Based Method
°F	degrees Fahrenheit
FEDS	Flexible Engine Diagnostics System
ft	feet
gpm	gallons per minute
GSE	ground support equipment
GTE	gas turbine engine
H	height
HEMTT	Heavy Expanded Mobility Tactical Truck
hp	horsepower
hr	hour
Hz	Hertz
in	inches
ISO	International Organization for Standards
kVA	kilovolt-amperes
L	length
lb_m	pounds mass

LC	labor cost
LCC	life cycle cost
min	minute
mL	milliliter
mph	miles per hour
ms	millisecond
MTBF	Mean Time Between Failure
NATO	North Atlantic Treaty Organization
NDI	non-developmental item
NPSHA	net positive suction head available
O&S	operation and support
OMB	Office of Management and Budget
OSHA	Occupational Safety and Health Administration
PARTS	mathematical variable, sum of spare parts cost and overhaul costs
PE	point estimate
PMO	Program Management Office
ppm	parts per million
psia	pounds (force) per square inch absolute
psig	pounds (force) per square inch gauge
RMF	repair maintenance factor
rpm	revolutions per minute
SCF	specific fuel consumption
TM	technical manual
US$	United States dollar
VAC	volts alternating current
VDC	volts direct current
W	width
WSARA	Weapon System Acquisition Reform Act
μ	micron or mean value
σ	standard deviation
ϕ	alternating current phase

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I. INTRODUCTION

A. BACKGROUND

The Army's Aviation Ground Power Unit (AGPU) was introduced in 1984 to support the AH-64A helicopter. This piece of equipment was designed to provide electrical, hydraulic, and pneumatic power to support the maintenance of the AH-64A, but it was adopted to support all of the Army's rotary aircraft fleet making the availability of the AGPU critical to the effectiveness of Army Aviation in the tactical environment. Early in the conflicts in Iraq and Afghanistan, the operational readiness of the AGPU was approaching levels as low as 5% (CSM Jay USA ret., personal communication, April 2, 2014). Spare parts became an issue and many began to conclude that this piece of equipment needed to be replaced soon, and perhaps, the current AGPU system architecture was no longer suitable for how the Army was using and maintaining the rotary wing fleet. Several initiatives were started with little success primarily due to inadequate application of systems engineering principles resulting in false starts and eventual cessation of the research and development efforts.

In the end, it was not the development of new systems or commercial-off-the-shelf (COTS) items that solved the operational readiness challenge; it was a service life extension program that did. Once refurbished units started arriving in theater, the operational readiness rate increased to 95% or better (CSM Jay USA ret., personal communication, April 2, 2014). This suggests that the current AGPU system architecture may not be obsolete for the current tactical environment. However, the dismal performance at the beginning of tactical operations does suggest that the AGPU may not be ideal for the total mission, which includes peacetime operations. One of the primary contributors to the initial poor performance of the AGPU in theater was the systems integration into the maintenance support infrastructure. System integration must consider the organizational construct that will support the system: parts, trained maintenance personnel, and processes at the user level.

Going forward, Army Aviation faces the prospect of replacing the AGPU with a new system or performing another major refurbishment. Goals of this new or improved system will surely include reducing operational costs and improving performance while increasing availability. To achieve this, "the architecture must have an operational context that goes beyond simply the realm of problem and system" (Maier and Rechtin 2009, 359). A successful system will not only have to optimize reliability, component cost, energy efficiency, and hazardous waste generation, it will also have to be supportable by the organizational construct to maintain availability through radical changes in mission profile, specifically making the transition from peacetime to tactical environments.

B. PURPOSE

This thesis examines the effectiveness of aviation ground power architectures comprised of various technologies to provide the functionality necessary to service the rotary winged aircraft fleet for the U.S. Army. Performance requirements are evaluated against total life cycle costs, system availability, and the ability of the organizations to adequately support the system.

C. RESEARCH QUESTIONS

What are the aviation ground power unit requirements?

What architectures will meet these requirements?

What are the life cycle costs of various architectures?

What features will allow the organization to best maintain the equipment?

Are there advantages to providing current functionality with multiple items?

D. BENEFITS OF STUDY

This study will benefit Army Aviation in effectively evaluating Aviation Ground Power Unit architectures that provide the functionality necessary to service aircraft. The study will inform materiel developers seeking to replace or modify the current AGPU on

design and requirement aspects that decrease costs, increase availability, and contribute to efficient supportability in the organizational constructs the equipment is to be used.

E. SCOPE AND METHODOLOGY

1. SCOPE

The focus of this thesis is the development of architectures that meet the current AGPU functions to provide propulsion, electric power, hydraulic power and pneumatic power. The total life cycle costs are evaluated based on unit procurement cost as well as on operation and maintenance costs. Availability is evaluated based on projected reliability and organization suitability is considered in the context of organizational training and logistics support.

2. METHODOLOGY

a. Perform literature review.

b. Review of current subsystem technology and COTS systems.

c. Research AGPU for performance parameters.

d. Research organizational support structure and constraints.

e. Interview subject matter experts on current system.

f. Evaluate subsystems' effectiveness in context of system and organizational architecture.

g. Recommend system architecture that poses the lowest life cycle cost in the current operational environment.

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II. REQUIREMENTS AND CAPABILITIES ANALYSIS

A. INTRODUCTION

As weapons systems capabilities evolve the requirements of the maintenance systems must also advance. Even though the basic requirements for electrical and hydraulic power will continue into the foreseeable future, how these services are supplied and the logistics footprint required to support them is certainly in flux. Fuel costs, the expense of hazardous waste disposal, inventory maintenance costs for support equipment spare parts, and the training cost associated with equipment all must be accounted for when evaluating systems prior to development and fielding.

An unintended negative consequence of complex systems is the adverse effect on support equipment. Funds and time allocated to train maintainers are highly constrained; as weapons systems become more complex, support equipment necessarily has less emphasis. As a result, reliance on manuals and corporate knowledge within units are very important to the development of requisite skills for proper operation and maintenance of support equipment. Therefore, future ground power unit architectures must not only provide capability with a smaller logistics footprint, they must also possess characteristics that lend themselves to supportability in a low formal training environment.

The primary piece of tactical support equipment in Army Aviation is the Aviation Ground Power Unit (AGPU). It provides the ground power requirements for the current U.S. Army and National Guard rotary wing aircraft fleet. Current rotary wing aircraft types supported include the AH-64, CH-47, OH-58, and UH-60. In addition, the C-12 fixed wing aircraft servicing may also be supported (Department of the Army 1986). As a fully contained self-propelled service cart, the AGPU provides both AC and DC electric power, hydraulic power, and pneumatic power. These various types of power are available individually or in combination with a JP-8 fueled gas turbine engine (GTE) serving as the power plant. System requirements stem from a composite of the aircraft supported along with general requirements applicable for aviation and general logistics support and operations.

B. FUNCTIONS

1. General

General requirements for support equipment encompass functions that enable the system to operate as a tactical asset with an aviation maintenance company. This specifically entails tactical environmental conditions, transportation, safety, and logistics footprint.

Operational environmental conditions for the AGPU range from -65 to 130°F at elevations from sea level to 10,000 ft on terrain up to 15° inclination (Department of the Army 2010). The system is expected to operate 500–1000 hours per year with a life of five years before depot level overhaul (CSM Jay USA ret., personal communication, April 2, 2014). The equipment may not pose a safety hazard to personnel or aircraft while functioning. Hot fluids under pressure combined with high voltage and high current pose an innate baseline operational risk that is mitigated through keeping the equipment in good working order and using proper procedures. Noise is an issue for this particular piece of equipment; extended shifts risk noise exposures above the OSHA limit of 90 dBA for continuous time-weighted eight-hour exposure (Proctor and Van Zandt 2008). A maintainer located at the operator panel will experience an acoustic environment of 90–93 dBA and 103–105 dBA at the hydraulic panel. A distance of 23 feet is required to attain a sound pressure level of 85 dBA (Department of the Army 2010). Exposures above the 85dBA threshold require the activity to institute processes to protect and monitor hearing (King 1996). Long-term exposure to high noise levels is detrimental to hearing and contributes to disability pay expenses, a cost that is difficult to incorporate into life cycle cost calculation of equipment. Lowering sound pressure levels created by the AGPU will do little to impact enterprise level costs; nonetheless, efforts should be made to avoid equipment contributing to these expenses when possible.

There are a number of characteristics required by the transportability function. MIL-STD-1366E (Department of Defense 2006) describes the transportation interface requirements for equipment in service. This piece of equipment must be transported via

truck, internal to aircraft (both fixed and rotary winged), and as an external load from rotary winged aircraft. These criteria define the limits of mass and overall dimension.

Shipping mass is the dry weight with a full complement of hydraulic fluid, engine oil, and a partial fuel load such that the system can be put into service once offloaded. The most stringent constraint for this requirement is the maximum radius mission for a UH-60A aircraft at 4000 ft and 95°F. For this mission, the UH-60A has a maximum external load capacity of 3,156 lb_m while the UH-60L has a capacity of 5173 lb_m (Department of Defense 2006). The absolute maximum mass for this item is governed by the crane on the Heavy Expanded Mobility Tactical Truck (HEMTT) M977A4 which has a lifting capacity of 4500 lb_m (Oshkosh Defense 2010). Shipping mass for the current system is 3620 lb_m (Department of the Army 2010) thus indicating potential for enhanced UH-60A mission capability if the system mass can be reduced. Therefore, the mass objective requirement is 3100 lb_m with a threshold requirement of 4500 lb_m.

Aircraft and ground transportation considerations govern overall dimensions. CH-47 equipment design limits are 80 inches wide by 72 inches high (Department of Defense 2006). Maximum length can be defined as 90 inches, the width of a HEMTT M977A4 minus 6 inches. During ingress and egress from the aircraft, the system must accommodate a 15° aircraft ramp angle and not contact the aircraft structure. The cargo ramp entrance height for the CH-47 is 78 inches (Department of Defense 2006). The current system has a height of 60 inches, a width of 58 inches, and length of 90 inches. Ingress/egress requirements are met with a wheelbase of 54 inches and minimum body clearance of 10.5 inches. System axel ground clearance is 7 inches. (Department of the Army 1986). The maximum envelope dimensions is 90x80x72 (LxWxH) in inches.

2. Propulsion

The propulsion function that dictates the system is self-propelled with a turning radius of 11 ft to enable maneuvering around aircraft on the flight line absent the need of another vehicle for positioning. A self-propelled flat surface velocity of up to three mph is required as well as a velocity 0.5 mph on a 26.8% grade (Department of the Army 1986). To accommodate flight line activities, the unit must be towable on improved and

unimproved surfaces with inclination of up to 15° in any axis with tow velocities of 20 mph on improved surfaces and 10 mph over unimproved surfaces. These requirements influence acceptable limits on center of gravity. A normally applied braking function is also required.

3. Electric Power

Electric power is the most used feature of the AGPU. Aircraft in the current fleet require two types of electrical power for maintenance, 28 VDC and 200Y/115VAC 3ϕ 400Hz. Voltage in direct current applications typically have a tolerance about them—for 28VDC nominal system this can be as much as ±4 volts. The AGPU is required to provide a continuous 350 Amps at a minimum voltage of 25 VDC. In addition, surge currents of 500 amps for 1 minute and 1000 amps for 5 seconds are required, but there are no minimum voltages required at these currents (Department of the Army 2010).

Rotary winged aircraft typically have maximum 400Hz power requirements that range from 15 to 17 kVA (Department of Defense 1993a). The AH-64 power demand is somewhat larger, approximately 32 kVA. The AGPU has a requirement to produce 48 kVA of 400 Hz power continuously. This provides for 115% of the maximum 400 Hz aircraft load and the ability to produce 250 amps of nominal 28 VDC simultaneously if required. A peak power load of 83 kVA is required for 30 seconds. Power quality requirements are governed by MIL-STD-704F (Department of Defense 1991). Aircraft voltage requirements are typically 108 to 118 VAC. However, external power being supplied to the aircraft must account for line losses within the aircraft; therefore, the interface voltage at the ground support equipment-aircraft interface is 113 to 118 VAC, allowing up to five volts of drop in the aircraft (Department of Defense 1993a).

4. Hydraulic Power

Performance requirements for hydraulic servicing of aircraft are pressure, flow rate, response time, fluid temperature to aircraft, and cleanliness. Table 1 provides a summary of the current requirements to support the hydraulic power function for aviation maintenance.

Table 1. Hydraulic System Requirements

Pressure (psig)[1]	Delivery Temperature (°F)[2]	Flow Rate (gpm)[1]	Suction Pressure (psia)[3]	Response Time (msec)[3]	Particulate (ISO 4406)[4]
500–3350	70–275	0–16.35	10.0 min	1000 max	17/15/12 min

[1]McCall 2009.
[2] Department of the Army 2010.
[3] U.S. Army Aviation and Missile Command 2010.
[4]Langhout 2014.

The lowest hydraulic operating pressure in the rotary-winged aircraft fleet is the OH-58A/C at 540 psig while the highest pressure required is for the CH-47 engine start procedure using a hydraulic servicing cart requiring a pressure of 3350 psig (McCall 2009). With the exception of the OH-58, army rotary winged aircraft have hydraulic systems that operate between 2900 and 3000 psig (McCall 2009). Because pressure surges must be kept to 135% of operating pressure, provisions must be made to accommodate the various aircraft operating pressures as well as maximum surge pressures (Department of Defense 1993b). It is an undesirable condition to have the aircraft vent hydraulic fluid during maintenance. Line losses on the pressure and return lines from the support equipment to the aircraft also must be considered. Currently, these line lengths would amount to about 40 psi drop total based on the distance between the ground support equipment (GSE) and the aircraft.

There are two types of temperature requirements—the previously discussed environmental temperatures and the temperature limits on the hydraulic fluid being delivered to the aircraft. Because of the diverse range of operating temperatures, weather conditions, and routing configurations, the current system was designed to deliver hydraulic fluid between 70–275°F (Department of the Army 2010). The upper limit is a typical upper operating temperature for engine and hydraulic oils, but it unclear what specifically drove the lower limit. Since no design documentation of the current system exists aside from the operating manuals and end item drawings, the lower limit will be assumed reasonable as it has been shown operationally viable.

Although the current system possesses a hydraulic pump capable of meeting the maximum flow requirements of 16.35 gpm, the installed configuration turns the pump shaft at 8000 rpm resulting in a maximum flow rate of 15.2 gpm (U.S. Army Aviation and Missile Command 2010). If maintenance actions require flows up to 16.35 gpm, then the current system cannot support those maintenance actions. Since the current system has been deployed for over 25 years, maintenance actions requiring 16.35 gpm are either not performed in the field, or there are methods to perform these tasks within the performance limits of the baseline system. So the question remains, what is the real flow requirement for the AGPU hydraulics system?

It was long thought, both on the flight line and in the engineering community, that the maximum flow for AGPU hydraulics was governed by the emergency start procedure for the CH-47. Where this thought originated is a mystery, but it was prevalent as requirements were being researched for AGPU alternatives. Unfortunately, documented values were not discovered. The procedure allows the AGPU hydraulic pump to be used in place of the aircraft system to start the CH-47 main engines.

In October of 2004, as Technical Chief of the Aviation Ground Support Equipment (AGSE) Program Management Office (PMO) this author and AGSE PMO staff in conjunction with the Aviation Engineering Directorate successfully demonstrated this function at Ft. Campbell, Kentucky. Flow rates measured from the CH-47 returning to the AGPU were approximately 12 gpm during the demonstration. Inspection of the data plates located on the aircraft hydraulic motors used to start the main engines themselves indicated the devices required 12.2 gpm (K. L. Alexandre, unpublished data). The maximum flow rate of the AGPU hydraulic pump happens to be 125% of this value. Therefore, it is reasonable to establish a minimum full flow pumping requirement of 15.2 gpm and an operating pressure between 500 and 3350 psig. These will be considered the threshold requirements.

The suction pressure requirement of 10 psia ensures that the current system will operate at the required elevation of 10,000 ft, at which the standard atmosphere is 10.11 psia (Pratt & Whitney 1990). This is an essential requirement for hydraulic systems with vented reservoirs. Little vertical space is available to utilize elevation head between

the reservoir and pump inlet. The only other option to provide sufficient pump inlet suction pressure is a separate boost pump located in the reservoir. Under some conditions, having a pump capable of operating at low suction pressures allows the system reservoir to be filled from bulk storage containers. Conditions are dictated by temperature, atmospheric pressure (elevation), and head loss due to lift and line loss.

Response times for typical variable displacement pumps are far less than the requirement for the current system. The reaction time is not as critical for maintenance as for flight, but the requirement should be at most 100 ms. Typical values are 25–75 ms (Eaton 2008), and it is not clear why *DMWR1–2910–300&P (U.S. Army Aviation and Missile Command 2010) allows a full second. This large value allows trade space for other hydraulic system architectures that may not have desirable performance characteristics.

The cleanliness requirement is based on the ISO 4406–1999 standard that specifies particle content in the fluid. Each number refers to a numeric range of particles/mL of fluid for a size class. The first number is the quantity range for particles/mL greater than 4μ; the second number is for particles/mL greater than 6μ; and the final number for particles/mL greater than 14μ. Operational fluid is deemed acceptable for Army rotary-wing aircraft at an ISO 4406 cleanliness level of 17/15/12 with up to 250 ppm of water (Langhout 2014). New fluid meeting MIL-PRF-83282D (Department of Defense 1997) is -/11/7 (Sauer-Danfoss 2010) in terms of ISO 4406–1999 with a maximum 100 ppm of water. This allows for some contamination during use. However, army rotary wing-aircraft hydraulic systems are not monitored for cleanliness leaving the ground support equipment the only indication of fluid suitability. Sampling of hydraulic ground servicing equipment is required after 50 hours of operation or every 30 days (Langhout 2014). The potential for regressive maintenance due to out-of-tolerance hydraulic fluid being introduced into an aircraft is significant. Therefore, having real-time analysis requirement of the fluid for particulate and water in future hydraulic ground support equipment is needed to mitigate unscheduled aircraft hydraulic system maintenance.

5. Pneumatic Power

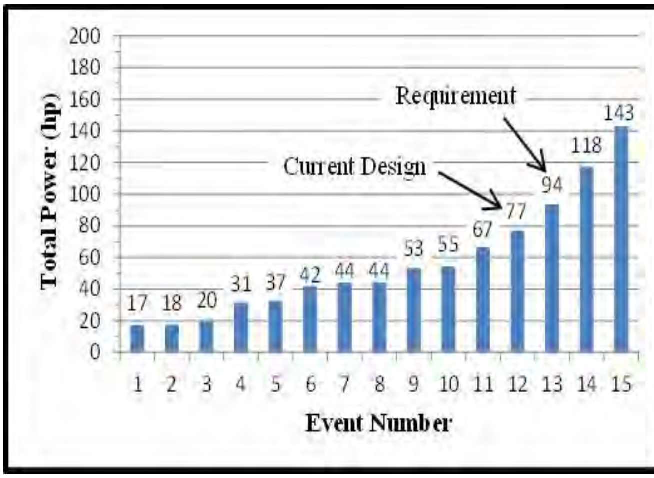
The T-700 series gas turbine engine is used by both the AH-64 and the UH-60. Engine start is initiated by expanding compressed air across a small air turbine, dubbed an air starter, to drive (spin up) the main engine for start. Pneumatic power for this operation is normally provided to the air starter from the aircraft auxiliary power unit (APU). If the APU or the 24 VDC system used to start the APU is inoperable, the ground support equipment must be capable of starting main engines of these aircraft. MIL-S-19557/11 (Department of Defense 1985) sets the upper temperature limit for air entering the air starter at 400°F with a maximum air flow rate of 26.3 lb_m/min at a maximum pressure of 45psia. There is no time duration or minimum flow rate specified.

In addition to having to supply compressed air to power the air starter, the AH-64D must be supplied with pressurized air during servicing for proper the hydraulic reservoir operation. This requires 26 psig at the aircraft hydraulic reservoir. The seal at this interface is notoriously leaky; therefore, a minimum flow rate of 2.0 lb_m/min (McCall 2010) is required by the GSE to ensure sufficient backpressure is maintained. The original AH-64 design used an air cycle machine fed by APU compressor bleed air to provide avionics cooling, so the AGPU was required to provide pneumatic power simultaneous with electrical and hydraulic power in order to satisfy all maintenance requirements for the AH-64. Updates to the AH-64 have replaced the air cycle machine environmental control with a refrigerant heat pump type system that has limited the pneumatic pressure function to starting the aircraft and providing pressure hydraulic system maintenance.

6. Power Plant

The power plant is a functional element characterized by derived requirements. How this function is employed has the greatest impact on overall design and performance. The power plant must be able to provide sufficient power to support simultaneous operations. Figure 1 provides required total power requirements for various maintenance scenarios. For a diesel engine this equates to shaft power, but for the GTE, the power is split between compressor pneumatic power and shaft power.

The AH-64 has the most power intensive maintenance, requiring pneumatic, hydraulic and electrical power simultaneously. In this configuration the maximum expected continuous load is defined by 2.0 lb_m/min of compressed air at 45 psia, 11.2 gpm of hydraulic fluid at 3000 psi, and 32 kVA of 400Hz 3ϕ electric power. This yields a continuous power plant requirement of 67 hp, requiring an engine that can produce 85 hp (125% of maximum continuous load estimate). The peak power requirement is estimated by using the peak electric power requirement of 83 kVA, which translates to a peak shaft power requirement of 91 hp for 30 seconds. The power plant in the baseline system is rated for continuous output of 62 hp (Department of the Army 2010); the gas turbine engine surge limit is at 77.5 hp, which is 125% above the continuous load (U.S. Army Aviation and Missile Command 2012). This means the baseline system cannot meet the peak power requirement.



Event	Description
1	5.6 gpm hydraulic @ 3000psi
2	350A @ 28VDC
3	Std Aircraft Electical
4	11.2 gpm hydraulic @ 3000psi
5	AH-64 Electric Pwr Only
6	Max Hydraulic Pwr
7	Max Pneumatic Pwr
8	1000A @ 25VDC
9	AH-64 Electric Pwr/ Single Hyd Sys
10	Max Rated Electric Pwr
11	AH-64 Electric Pwr/ Dual Hyd Sys
12	125% TM Max Engine Pwr
13	Peak Electrical Pwr RQMT
14	125% Peak Electric Engine Pwr
15	Max Electric Pwr/Max Hyd Pwr/Max Pneumatic Pwr

Figure 1. Shaft Power Requirements

Hydraulic pump efficiency is assumed to be 70% (Eaton 2008) and pneumatic compression is assumed to have an efficiency of 75% (Boyce 2002). Electrical generation equipment is assumed to have an efficiency of 95% for AC power and 76% for DC power (Baldor Electric Company 2014; U.S. Army Aviation and Missile Command 2007).

C. CHAPTER SUMMARY

This chapter examined requirements germane to maintaining Army rotary winged aircraft in a tactical environment. The requirements discussed pertain primarily to those that specifically affect the power for aircraft maintenance, logistics, and human interfaces that can be influenced by variations in system architecture. For instance, the sound pressure produced by the power plant may require increased system mass and volume to rival the performance of a different technology possessed by a competing architecture. Impacts on operations and long-term compensation are areas that can be assessed in design suitability comparisons. Other human factors, such as cold weather gear interface compatibility are not considered because there is little relevant design space available to differentiate architectures—one must assume each offering is compliant.

III. ARCHITECTURES

A. INTRODUCTION

Useful architectures that provide the required functionality are limited. The area where interesting distinctions can be achieved is how the various functions are coupled to the power plant, a strong function of the power plant itself. For this study, only two types of power plants will be considered: gas turbine and diesel engines. Many types of power plants have been studied to satisfy mobile power requirements for tactical systems, and these two technologies consistently emerge from the trade studies as the preferred choices. Light fuel power plants such as the Otto and Wankel engines are excluded from consideration due to the JP-8 fuel requirement. The attractive power to weight ratio of the Wankel did lead the army to develop a prototype heavy fuel Wankel engine. Testing in 2005 demonstrated a marked decrease in shaft power as well as other performance issues that resulted in suspension of research efforts by the ground support community (K. L. Alexandre, unpublished data).

Two studies conducted by the army, one by the Aviation Engineering Directorate and the other by the AGSE PMO, both determined that a fully distributed system architecture was not preferred for the tactical system. This author participated in the AGSE PMO study and engaged with Jerome Smith who performed the other. The presumption of the distributed architecture was that having separate pieces of equipment for each function could achieve a cost advantage by minimizing the total amount of equipment enterprise wide. Issuing of the equipment for the various maintenance organizations would be based on usage rates dictated by maintenance actions within those organizations.

For example, the OH-58 uses the ground support equipment primarily for electric power, yet the units are carrying the overhead associated with hydraulic and pneumatic power. For the aviation enterprise, distributed functionality would decrease the total quantity of hydraulic pumps and the associated cost savings could be realized. Unfortunately, the increase in mass and volume of the alternatives overwhelmed the

quantity reduction of end items. An AH-64 unit would potentially have thrice the equipment to achieve the same functionality. The agglomerated equipment would have a larger logistics foot print than the architecture it was intended to replace, thus the decision was made to keep the multifunctional architecture.

Within the multifunctional architectures there are variations that can be explored for best life cycle cost. This is a result of the power plants having unique sets of functionality as well as various means to transfer power to the functional elements. This report examines each function to determine the best technology options to meet the functional requirements. To limit the permutations of top level architectures, the same functional components will be used with both power plant options. This approach yields four unique systems for examination: GTE with shaft driven hydraulics, GTE with electric drive hydraulics, diesel with shaft drive hydraulics and pneumatics, and diesel with shaft drive pneumatics with electric drive hydraulics. Top-level system architectures are depicted in Figure 2.

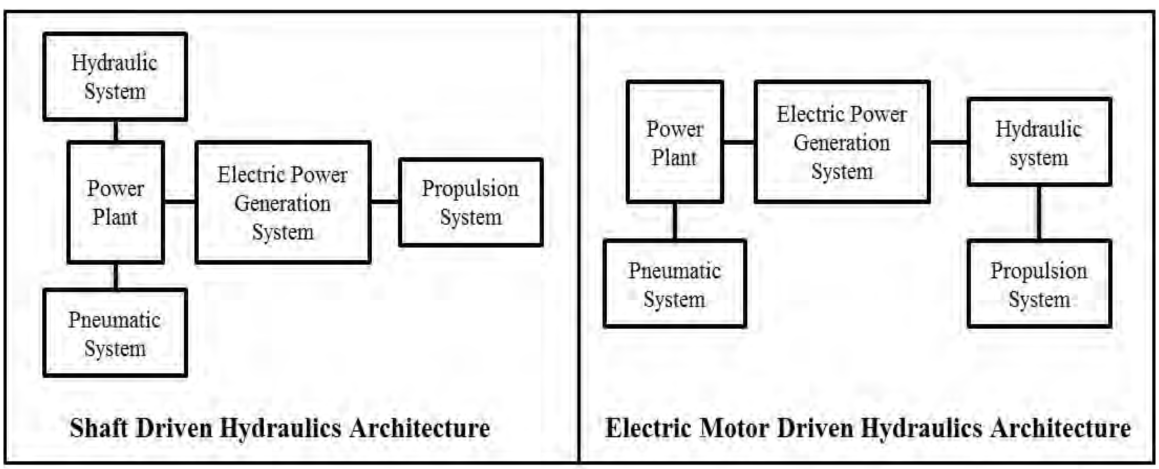


Figure 2. Top-Level System Architectures

For this thesis, the systems with electric drive hydraulics will use the hydraulic system for the propulsion function, and the systems with direct drive hydraulic pumps will utilize the heritage type electric motor drive propulsion. The AGPU, having a gas

turbine engine and shaft driven hydraulics, is the current baseline and will provide the basis of comparison for the other three architectures.

B. PROPULSION SYSTEM

The propulsion system currently is comprised of a three hp 28 VDC electric motor that drives a differential on the rear axle. Powering the propulsion system with the direct current bus allows the battery pack to provide power for this function when the power plant is not operating. This allows the cart to be maneuvered in enclosed areas, such as a storage hangar, or adjacent to aircraft when communicating easily with others is critical. The electric motor and transmission used in the baseline system has a mass of about 380 lb_m (CSM Joseph Jay USA ret., personal communication, April 2, 2014).

For the electric motor driven hydraulic system architecture, an option would be to power the main hydraulic pump from the battery pack by inverting 24 VDC battery power to 200/115 VAC 3ϕ 400 Hz power, and using the hydraulic pump to power a hydraulic motor at the rear differential. An inverter with the capability to drive an 8 gpm pump would have a mass of approximately 180 lb_m (PowerStream Technology 2014) and an envelope of 15x19x42 inches (LxWxH). This assumes a hydraulic system consists of two 8 gpm pumps in parallel with only one of the two being utilized for propulsion.

Using the baseline differential and axle system and replacing the traction motor (140 lb_m) with an inverter (180 lb_m) and hydraulic motor (15 lb_m) will increase the propulsion system mass by approximately 40% to 195 lb_m. In order to achieve a mass advantage with a hydraulic propulsion system the entire differential and axle would need to be replaced with independent hydraulic motor drives at each rear wheel. This should bring the total system mass down to the 150 lb_m range; unfortunately, this mass reduction requires a significantly more complex propulsion control system. An inverter is also required for this option. The volume required will also be reduced compensating somewhat for the inverter envelope.

In summary, there are three propulsion system options with the following masses:

1) Baseline – transmission with electric motor drive 380 lb_m.

2) Hydraulics Option A – transmission with hydraulic motor drive 455 lb_m.

3) Hydraulics Option B – independent wheel hydraulic motor drive 330 lb_m.

Hydraulic Option B is the preferred propulsion alternative to the baseline for architectures having electrical motor driven hydraulic pumps.

C. ELECTRIC POWER SYSTEM

The electric power generation system utilizes a precise generator to produce 200/115 VAC 3ϕ 400 Hz power. In the precise generator scheme, the power plant is controlled at a fixed speed to produce the desired power frequency based on the number of the poles contained in the generator. Generators that produce 400 Hz power are commercially available with 6 to 24 poles; this sets the engine shaft speed trade space between 8000 rpm to 2000 rpm.

Creating commercial power (460/266VAC 3ϕ 60Hz) instead of aircraft power (200/115VAC 3ϕ 400Hz) would enable use of less expensive generators and electric drive motors that operate at lower shaft frequencies. It also allows standard hangar shore power (Ferriter 2012) to be easily utilized in lieu of the power plant generated electricity. This would provide value to the maintenance unit by making the GSE available for hangar operations, enabling the same equipment to be used in the hangar as would be used in a tactical environment. In addition, all the cables, hoses, and other items that experience degradation over time would have more visibility within the unit so that these items are unlikely to become unserviceable.

For the specified power levels the mass penalty to convert 460/266VAC 3ϕ 60Hz to 200/115VAC 3ϕ 400Hz power for aircraft maintenance would be approximately 1500 lb_m. This increase in mass cannot be accommodated by the tactical system; therefore, the onboard electrical generation system must produce 200/115VAC 3ϕ 400Hz power and all electric drive motors operate on 400Hz power or 28VDC power. To use hangar power, an architecture featuring electric powered hydraulics would be required

with the application of a non-tactical power conversion kit. Operators would remove this kit prior to tactical deployments.

Direct current power is produced by a transformer rectifier using the 200/115 VAC 3ϕ 400 Hz as an input and producing 28VDC as an output. The requirement for 30 amps of standard 120 VAC 1ϕ 60 Hz power and is supplied off the 28VDC power system by a COTS inverter. The 120 VAC 1ϕ 60 Hz power system supplies power to hand tools, lights, and other auxiliary equipment and is not intended to power any GSE functionality. AGPU components for these functions will be assumed for all architectures discussed herein.

D. HYDRAULIC POWER SYSTEM

1. Hydraulic System Architecture

The hydraulic system consists of lines, accumulators, valves, heat exchanger, manifolds, reservoir, and pump. Of these components the primary cost driver and highest non-consumable item is the pump. Failure of non-consumable items (lines, accumulators, valves, heat exchangers, manifolds, and reservoir) is rare. Therefore, the sole focus of configuration changes to the hydraulic system will be related to the “pump hydraulic fluid” function.

It is important to note that when considering component selection the hydraulic system is classified as an intermittent system (Miller 1987) as opposed to a continuous industrial system. Periods of operation similar to a continuous system may be experienced, but the aggregate operating environment is less stressing than a continuously operating system allowing use of lower mass and less expensive components while keeping reasonable reliability.

2. Hydraulic Pumps

A myriad of pumps are available within the family of positive displacement pumps suited for hydraulics systems. A number of influences must be considered, from the type of systems the GSE must support to using disposable vise repairable. Consumer use of plunger pumps has influenced the discussion of disposable hydraulic pump use in

aircraft maintenance. These devices provide high pressure intermittent flow with seemingly good reliability and are reasonably inexpensive. A 2006 cost evaluation performed by the author of disposable plunger pumps and depot repair of the AGPU pump determined that the procurement costs of the disposable pump were significantly less than the cost of depot repairs of the in-service AGPU hydraulic pump (K.L. Alexandre, unpublished data). Though the cost of the disposable pump was less than the overhaul costs of the repairable pump, the significant modifications to the existing system eliminated the disposable pump from further consideration. A survey conducted by AGSE PMO in conjunction with DLA of manufacturers found that there were no repairable COTS pumps on the market that met both the performance and interface requirements of the AGPU (K. L. Alexandre, unpublished data).

The previous studies that investigated disposable and repairable COTS replacements for the AGPU pump did not examine design aspects beyond the interfaces, operating pressure, and flow rate. If the desired change were viable, additional system requirements would have been discovered. Typical aviation hydraulic pumps have functionality that inexpensive pumps do not have; therefore, the system design must account for this functionality with other components. Two required pump functions that can be performed with components, other than the primary pump, are: “provide ample net positive suction head available (NPSHA)” and “provide variable flow.”

a. Net Positive Suction Head Available

NPSHA is the total suction pressure less the vapor pressure of the fluid at operating temperature, or the “pressure above the vapor pressure required to fill the cylinder volume with fluid during suction” (Miller 1987, 53). The term total pressure is the sum of the static and dynamic pressures and for positive displacement pumps the dynamic pressure is the more important component of the suction head. Sufficient dynamic head must be available to move fluid into the cylinder. As this occurs, the interaction between the fluid and the piston within the cylinder is critical to pump performance, and if not managed properly, will result in cavitation and excess pump

vibration. These are the two dominant adverse effects on NPSHA in positive displacement pumps (Wachel and Szenasi 1986).

As the piston retracts in the cylinder, fluid in proximity of the retracting surface experiences an increase in specific volume that necessarily results in a simultaneous decrease in local pressure in accordance with the thermodynamic equations of state for the fluid (Smith and Van Ness 1975). Note that for liquids, very small changes in specific volume create large changes in pressure. If the local pressure drops below the vapor pressure of the fluid, a vapor “bubble” will form and collapse resulting in a phenomenon known as cavitation. As the “bubbles” collapse, they transfer their energy to a very small area. If this energy is transferred to a pump surface, it will cause pitting and erosion of the surface resulting in degraded performance and decreased life. To further complicate the matter, vapor pressure is a function of fluid temperature. The higher the fluid temperature the higher the vapor pressure, meaning that as operating temperatures increase, local pressure must remain high to stay above the vapor pressure. This increases the importance of properly managing the dynamic head at the pump inlet for high temperatures experienced in tactical environments.

Also affecting the dynamic head are pressure waves that cause fluid accelerations in opposition to the normal flow. The pressure waves are caused by the moving parts of the pump interacting with the fluid, similar to water hammer in piping systems. The moving parts of the pump have a different velocity with respect to the fluid. The fluid is decelerated as it contacts these surfaces and pressure waves are created that travel against the inlet flow adversely affecting the dynamic head. These pressure waves can create excessive vibrations if design precautions are not exercised.

The primary method to control these adverse effects is to provide centrifugal suction boost pump and a suction stabilizer (Miller 1987). Placing a centrifugal pump at the inlet actively aids the maintenance of dynamic head while the suction stabilizer acts to diminish the effects of the reflected pressure waves on the fluid entering the pump.

Variable displacement hydraulic pumps used in aviation commonly provide an integral centrifugal pump suction boost section, dampened pistons, and piston inlet

design that minimizes pressure waves caused by piston accelerations. When integral suction boost is not utilized, a centrifugal pump is located in the supply reservoir.

b. Flow Control

As alluded to earlier, aviation hydraulic pumps are typically variable displacement pumps, adjusting flow to maintain a set pressure. Full flow occurs when the pistons are allowed to attain maximum stroke. When there is no demand for flow, a swashplate changes position and minimizes the piston stroke to produce only enough flow to meet the minimum cooling and lubrication needs of the pump. This is referred to as the case drain flow. Because this low flow rate is occurring at pressure, this state of operation induces wear that reduces pump life. Variable capacity control methods using a constant displacement pump are drive speed control, suction valve unloading, and bypass of excess flow (Smith 1986).

Constant flow pumps are typically the least expensive of the positive displacement pumps and therefore they appear attractive from a cost perspective. The flow of these pumps is solely a function of shaft frequency. Controlling the flow by varying the shaft speed has the advantage of increased reliability. Since the pump will rarely operate at full flow for extended periods of time, the average shaft speed is greatly reduced thereby extending the life of the rotating group and thus the pump. Erikson, Sabini, and Stavale (n.d.) report abrasion from contaminants, degradation of oil and oil additives, seal wear, and bearing temperatures all increase dramatically when shaft speeds exceed 50% of the maximum design limit. The data also show that when the pump speed is reduced by half the reliability can increase by a factor of five (Erikson, Sabini, and Stavale n.d.). Full range control authority with respect to flow can be achieved by varying the pump shaft speed electrically or mechanically.

The most popular and efficient approach flow control method is variable frequency motor control of which there are at least a dozen methods utilizing both alternating and direct current motors (Industrial Technologies Program 2004). Relative efficiencies around 90% can be achieved for almost all shaft speeds (Smith 1986); the relative efficiency being defined as the system efficiency at a flow rate relative to the

system efficiency at full flow. However, this method of control can only be applied to hydraulic systems using an electric motor to drive the hydraulic pump. There are other considerations that make this method of control less desirable for a GSE application: the electric motors with stator mounted cooling fans may not provide adequate air flow at the low speeds; effects of motor torque at low shaft speeds may adversely affect electronic components; and shaft speed variations may create adverse vibrational resonance in the pump (Industrial Technologies Program 2004). In addition, stray current can be carried through the bearings and shaft of the pump resulting in micro arcs that can create failure initiation points on critical surfaces. These types of failures can be particularly vexing to resolve in mobile equipment where variations in grounding, operation, and environment are not consistent. These factors contribute to addition failure modes for the pump and the potential to reduce pump reliability compared to the same pump in a fixed installation.

Mechanically variable flow control can be accomplished using fluid or magnetic coupling devices. These devices can achieve full flow range control authority and operate with a constant shaft speed input enabling use with either electric motor drives or power plant accessory drives. All these coupling devices have very poor low shaft speed relative efficiencies (Smith 1986). These devices replace the swashplate function in a variable displacement pump. The negative aspects to these items are the mass and volume required for integration as well as low relative efficiency at low pump shaft speeds. Pressure waves reflected through the system by the pump can also affect controller performance of these devices (Yeaple 1995).

Unloading valves are used to remove the pressure load from a pump when the desired system pressure is reached and to divert the full flow back to the reservoir (Goodwin 1963). When the system unloads, the pump is essentially working against reservoir pressure, so time under this operating condition is not counted against pump life (Smith 1986) even though the pump is turning at full speed. An unloading valve provides the system with no flow or full pump flow. Demands for hydraulic power during aircraft maintenance can be best characterized as intermittent flows of short duration, a condition that can impede smooth system operation when an unloading valve is used for flow

control. Anecdotal evidence from author discussions with maintainers suggests that many hydraulic maintenance operations can be successfully accomplished with equipment providing only 3 gpm of flow, suggesting that on/off control of a 16 gpm pump could prove to be problematic. Having two separate 8 gpm systems operating in tandem for high flow rate operations would make the situation more tenable, but it would not alleviate the potential for rapid exercise of the unloading valve making the unloading valve a reliability concern. Special care would have to be taken in the design to produce an acceptable system pressure profile and system response time. Selection of on/off control is typically for applications such as presses (Miller 1987) and pressure washers.

The last type of flow control for consideration is bypass flow. In this control method, the pump operates at system pressure and maximum expected flow. Excess flow is diverted back to the reservoir. This approach uses the most energy and produces the maximum wear on valves and the pump. This type of system is extremely simple, inexpensive, and avoids system fluctuations expected with an unloading valve.

Of the three flow control options for use with constant displacement pumps, the fluid or magnetic coupling devices are the most attractive. Variable frequency control cannot be used with shaft driven hydraulic systems and present unique issues that make it problematic in the tactical environment. Use of unloading valves is not the correct application for this system and by-pass control uses too much power and causes the most pump wear of any of the flow control options.

3. Hydraulic System Comparison

a. Variable Displacement Versus Constant Displacement

Variable displacement pumps have the benefit of minimizing the total number of components and simplifying installation checks after replacement. Functions that are combined in the pump are tested as a unit prior to being shipped to the field. Installations issues that may result from integrating the pump with a variable flow control device are not present.

This author and CSM Jay conducted a study of Code F (parts deemed failed but repairable by a field unit using technical manual criteria) variable displacement hydraulic

pumps awaiting depot rework and found that a full third of the pumps failed by the maintainers passed the acceptance test procedure (ATP) before any rework was performed by the depot. A little more than one third had properly functioning rotating groups but failed compensator motors (small electric motor used to adjust pressure). The remainder had failures of the rotating group itself (K. L. Alexandre and J. C. Jay, unpublished data). In this work, instances of excessive wear and catastrophic failures of the rotating group were attributed to fluid contamination and fatigue induced by improper system operation. This is an indication that reliability issues would not necessarily be alleviated by changing the style of pump and may in fact suggest that adding components and integration complexity would result in lower system operational readiness rates due to increased replacement of serviceable components. The more components contribute to a malfunction, the greater the opportunity for unwarranted removal and replacement actions. Evaluation of the variable displacement pump shows it to be well suited for this application.

Both fluid and magnetic coupling devices size and mass are comparable to that of the pumps themselves; the net result is performing the "variable flow function" with higher mass and lower reliability. Assuming that fluid coupling devices, constant displacement pumps, and variable displacement pumps have similar reliability, placing components with the same failures modes and failure rates in series reduces the overall reliability. Even if the two devices operating in series are able to achieve the same reliability as the variable displacement pump, the mass and volume of the mass and volume of the combination negates any advantage in separating the "variable flow function" and "hydraulic power function" into different components. A variable displacement pump is the better selection for this application.

b. COTS versus Military Qualified

Substitution of COTS pumps for military qualified variable displacement pumps is of interest as a potential cost savings. Commercial pumps are typically rated only for temperatures between -10°F and 220°F and not -65°F and 275°F. A survey of commercial variable displacement pumps also revealed consistently higher inlet pressure

requirements, ranging from 12.3 psia to 14.9 psia versus the 10.0 psia requirement for the current AGPU pump. COTS pump masses ranged from 22 lb_m to 33 lb_m while the current military qualified pump is only 15 lb_m. In order to meet the requirements utilizing a similar COTS pump, an inlet boost pump or pressurized reservoir, additional heat exchangers, and additional controls would be required.

Two approaches address these requirements deficiencies. The first is to simply change the tactical systems requirements to align with COTS hardware capability. Mining and oil companies operate in very harsh conditions and the environmental requirements for COTS equipment are sufficient for those industries. If tactical environments were found to cause degradation in reliability, testing to develop reliability data for COTS pumps at tactical environmental conditions would be required to establish proper inventory levels.

Using kits designed for harsh environments is the other option. A kit is an approved set of equipment that changes the configuration of an existing system in service; a “B” kit is applied in the field while an “A” kit is applied at a depot or approved maintenance organization. Kits would be designed to protect the COTS items from the extreme environmental conditions and for ready application to meet mission requirements. The non-recurring engineering to establish interfaces to add the required functionality would increase system procurement costs. Logistically there is also a recurring cost to provide and manage the kits even though these would be very low demand rate items under normal circumstances.

COTS pumps are also subject to the shrinking manufacturing base and may change configuration without notice. This requires that the program management office establish a very rigorous configuration management process. Without a mechanism to detect configuration changes, pumps procured under technically acceptable lowest cost criteria may not be suitable for use. It is not uncommon for items with the same part number to change significantly in mass, volume, configuration, or any combination thereof. The recurring cost of maintaining source control drawings or vendor control drawings to manage consumable components must be weighed against potential savings gained by use of COTS equipment. The central procurement agency for consumable parts

might not consult these documents even if the program management office establishes the means for configuration management. In the current system, the control drawings will stop procurement or prevent populating the logistics system with procured items while the procurement official contacts the program management office about discrepancies between the end item and the requirements documents.

c. Summary

The highest hydraulic system availability will be realized by using two variable displacement pumps operating in parallel at a reduced speed with respect to the maximum rated capacity. Since one 8gpm pump can perform many hydraulic maintenance operations, pump life would be further extended. Given that the military qualified pump is over twice the cost of a comparable COTS pump, it seems reasonable to opt for extreme weather kits in conjunction with two 8 gpm COTS pumps in parallel for the preferred solution. Using this configuration will necessitate a suction boost pump as part of the base configuration enabling use of vented reservoir which is less expensive and lower maintenance than a pressurized reservoir of equal capacity. The disadvantage is a mass increase of 30 lbm to 50 lbm assuming the drive does not require additional gear reduction.

E. PNEUMATIC POWER SYSTEM

1. Compressor Types

There are two broad categories of compressors: positive displacement and aerodynamic. Of the positive displacement compressors, reciprocating, rotary screw, and sliding vane technologies are suited to the flow and pressure requirements (Khan 1984).

Based on adiabatic head and specific speed calculations at the flow and pressure ratios under consideration, Figure 3 implies the sliding vane compressors is an attractive alternative. However, these compressors generate wear debris and require active cooling (Yeaple 1995). Though the compressors are somewhat compact, the auxiliary equipment required is heavy and voluminous. A COTS solution for this application is in excess of 500 lb_m (Gardner Denver 2009).

Of the positive displacement class of compressor the single stage piston and the rotary screw compressors are best suited. The rotary screw machines are close tolerance complicated machines with high relative procurement and maintenance costs while also presenting an oil contamination risk (Yeaple 1995). High mass and COTS solutions in the flow range are also issues. For this application the reciprocating compressor appears to be the best alternative of the positive displacement machines.

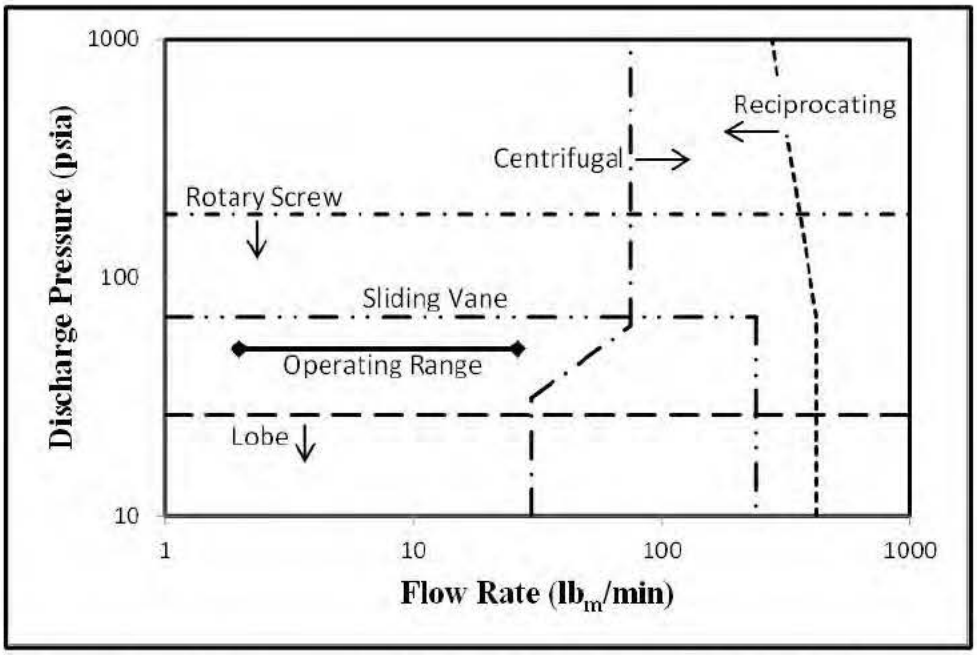


Figure 3. Compressor Technology Suitability (after Khan 1984)

Of the aerodynamic compressors, the centrifugal compressor is the only possible candidate. However, it is not ideal from a pressure and flow perspective as shown in Figure 3, neither is it ideal for the desired shaft speeds associated with a typical diesel engine. Centrifugal compressors operate best at specific speeds between 60 and 1500 (Boyce 2002); 30 to 3000 is acceptable (Khan 1984). The requirement for an order of magnitude variation in flow rate at a constant pressure ratio in this application is problematic for the technology.

The reciprocating and centrifugal compressors will be examined for use with the diesel engine architectures.

2. Diesel Engine Pneumatic Power Architecture

The advantage of this architecture is that the compressor does not have to produce compressed air when there is no demand—the device can be clutched and only operate when pneumatic power is required. Pneumatic power usage is predominantly associated with AH-64 hydraulic system maintenance that requires a low mass flow rate and is used in combination with other power demands. The high flow demand is used only to start or de-ice aircraft, not in conjunction with other functions. If the maximum required pneumatic power is being generated during normal maintenance activities it would unnecessarily cause the power plant to be oversized, thus being necessary to have a pneumatic power system with the capability to deliver the pneumatic power required for the specific operation.

Several factors must be considered when choosing components and a pneumatic power configuration for use with a diesel engine. Integration of the pneumatic power function and the diesel engine power plant beyond a simple shaft interface must be avoided because aviation units are not allocated personnel with training to perform maintenance on diesel engines. The aviation activity is not a large user of diesel engines. Therefore, it is imperative to maximize diesel engine configuration commonality by not introducing unique aviation variants to the supply system while also to keeping the pneumatic power interface simple enough such that aviation personnel can perform the required remove and replace maintenance operations. This limits the packaging and integration trade space barring potential features that could lend themselves to decreased mass or increased efficiency.

a. Positive Displacement Compressors

To address the full range of the flow requirement utilizing a positive displacement compressor, one must develop a method to vary capacity. The preferred method to vary capacity is speed control of the drive motor (Yeaple 1995). Since the power plant is controlled to a specific shaft frequency to support the electric power function, a simple system would provide two discrete shaft frequencies corresponding to the desired flow rates through independent drive pads or a selectable gearbox. Another approach to vary

flow is with suction unloading or changing the effective volume of the cylinder while using a constant shaft speed. The latter is accomplished by controlling a valve that connects the piston to the additional volume thereby lowering the cylinder pressure achieved during the piston stroke. Either approach is reasonable.

Perhaps the biggest disadvantage, as with the other positive displacement alternatives, is the mass of the machines. Estimates for the high flow requirement based on the mass correlation depicted in Figure 4 indicate the expected compressor mass at approximately 600 lbm. This means that there is a tremendous weight penalty (16% of current system shipping mass) attributable to the least used function of the ground support equipment. The low flow condition may be met with a machine mass of approximately 60 lbm.

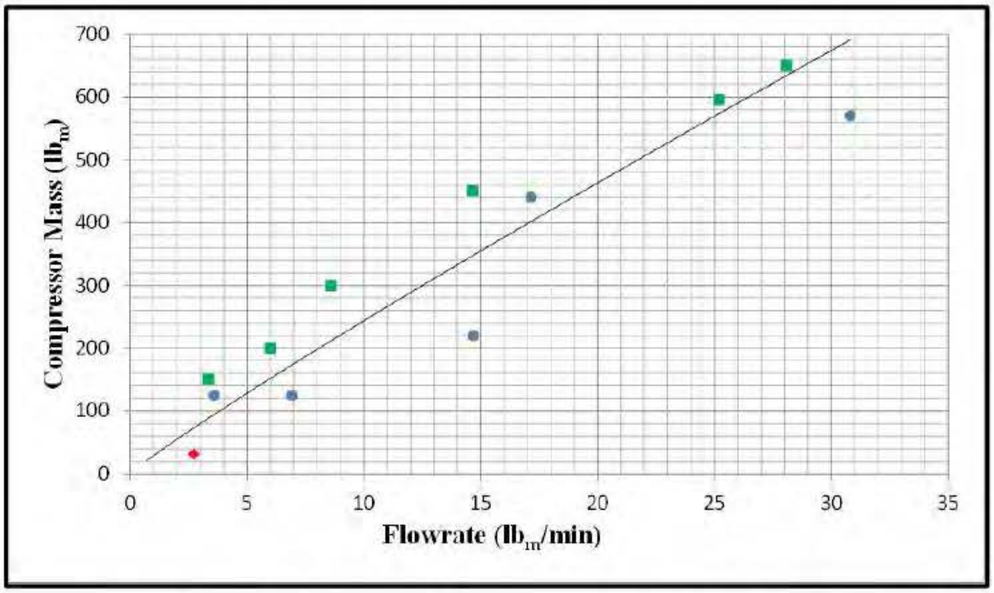


Figure 4. Reciprocating Compressor Mass Correlation

b. Dynamic Compressors

"Flowrate, efficiency, and pressure rise within the compressor are the three most important parameters used in defining the performance of a compressor and its selection" (Boyce 1993, 161) and evaluation of these parameters indicate direct drive centrifugal compressors are not ideal for this application. Shaft frequencies for centrifugal

compressors range between 3000 and 200,000 rpm. Operating this type of compressor with a gearbox driven by a diesel engine requires minimizing the compressor shaft frequency. Depending on the engine and paired precise generator, the engine shaft frequency could range from 2000 to 4000 rpm. Using a maximum gearbox ratio of 20:1 yields a range of available compressor shaft frequencies between 40,000 and 80,000 rpm.

As compressor shaft frequency decreases, the diameter of the machine increases for a given flow rate. The trade space is best defined using dimensionless parameters of specific speed and specific diameter. Practical limits of operation for radial centrifugal compressors dictate specific speeds between 30 and 70 depending on the diesel engine speed. The trade space for the specific diameter based on the operational map developed by Balje (1962) is between 1.9 and 4.5; this precludes use of automotive turbocharger compressors and industrial compressors used in wastewater treatment applications. Automotive compressors require speeds between 140,000 and 160,000 rpm to achieve desired pressure and flow while the slower turning single stage industrial counter parts are challenged by the desired pressure ratio. Nonetheless, Figure 5 was derived for the worst-case requirement, 26.3 lb_m/min, 130°F, and 10,000 feet altitude, to provide a method to estimate size and weight of the required machine based on the relevant specific speeds. If the highest practical direct drive shaft speed of 80,000 rpm is utilized the compressor impeller diameter is approximately 9 inches in diameter with a mass around 12 lb_m. Envelope size estimate for the gearbox and compressor is approximately four cubic feet, 24x12x24 inches (LxWxH), with a mass of 150 lb_m based on similarity to industrial machines.

The high flow rate condition may be satisfied using the radial centrifugal compressor, but the full range of flows required cannot be addressed with this machine alone. When maximum pneumatic power is required, as with aircraft start and de-icing operations, there are no other demands on the power plant. At the low flow condition electric and hydraulic power are required and it is not possible to support these loads when the maximum pneumatic power is being generated. This implies that to meet the full range of requirements a centrifugal compressor must be matched with a small

reciprocating compressor or the centrifugal compressor must be paired with a turbine to scavenge power from the excess air produced.

From a mass perspective, these two options are somewhat comparable for direct drive applications. As previously stated, the small compressor would be about 60 lb_m. Pairing a turbine with the centrifugal compressor would increase the pneumatic system mass by about 40 lb_m. Adding a turbine increases complexity and power consumption because the turbine will recover only about 85% of the power from the excess air produced. A direct driven high flow centrifugal compressor with a direct drive low flow rate reciprocating compressor would have a mass of approximately 210 lb_m.

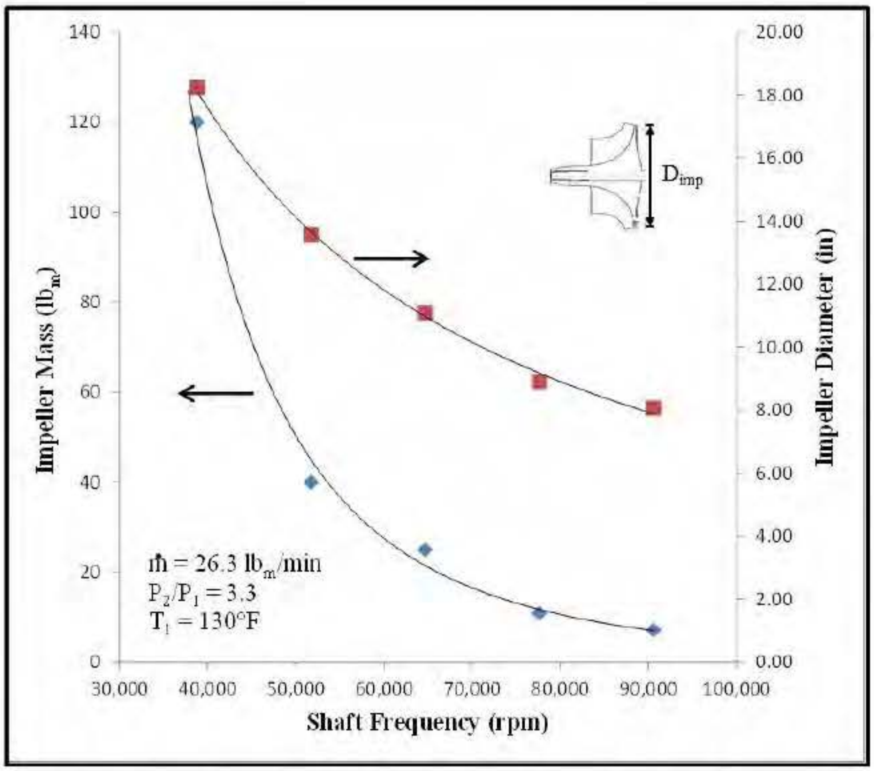


Figure 5. Impeller Characteristics (after Balje 1962; Xu and Amano 2012)

To open the trade space to higher turbine shaft frequencies, the architecture can be altered to drive a centrifugal compressor with an electric motor similar to those being proposed for the hydraulic system. By using a 200 VAC 3ϕ 400 Hz motor to drive the compressor, shaft speeds of 180,000 rpm could be reached using a 15:1 step-up gearbox

allowing small automotive compressors into the design space. The mass of the electric motor and gearbox is approximately 135 lb_m; an additional 10 lb_m for the small centrifugal compressor yields a total mass of 145 lb_m. This design is slightly lighter and provides more flexibility in packaging that could allow for a simple interface to waste heat. Interfacing to power plant waste heat would enable the turbine to support the low flow rate condition more efficiently. This option has the benefit of trading weight reduction for higher system complexity while staying within the maintenance constraints. The magnitude of the mass trade space is about 40 lb_m when compared to using a small reciprocating compressor. Decreased reliability is the primary negative to using a smaller turbine at significantly higher shaft speeds as failure rates increase with increasing shaft speed. An electrically driven high flow centrifugal compressor with a direct drive low flow rate reciprocating compressor would have a mass of approximately 205 lb_m.

c. *Diesel Engine Pneumatic Power Architecture Summary*

Because there is not a dominant architecture, a weighting method of system attributes is used to select between alternatives. The attributes examined are mass, efficiency, complexity, and required maintenance. Mass and efficiency relate directly to field operations while complexity and maintenance lend themselves more toward initial and operational cost respectively. Table 2 shows that the preferred method to meet the pneumatic power requirement is using two separate components, one for the high flow and another for the low flow rate requirements.

Table 2. Diesel Engine Pneumatic Power Alternatives

Criterion	Weight	ALTERNATIVES					
		Large Reciprocating Piston Compressor		Centrifugal Compressor With Scavenging Turbine		Centrifugal Compressor and Small Reciprocating Piston Compressor	
(parameter:rating)		Rating	Score	Rating	Score	Rating	Score
Efficiency ↑↑	0.35	7	2.45	4	1.40	6	2.10
Mass ↓↑	0.35	2	0.70	5	1.75	5	1.75
Complexity ↓↑	0.15	4	0.60	6	0.90	3	0.45
Maintenance ↓↑	0.15	5	0.75	5	0.75	4	0.60
Composite Score	Better ↑		4.50		4.80		4.90

The evaluation shows the efficiency gained by using components best suited for the flow rate combined with the least amount of integration complexity predominated. No differentiation was made between the electrically driven and direct drive centrifugal compressor since in terms of mass they are essentially equivalent. The electrically driven system affords the greater flexibility in packaging while also providing the option to reduce mass if warranted and therefore would be the preferred. Thus to add the pneumatic power capability to a diesel power plant requires an additional 210 lb_m and 5 ft^3 of space.

3. Gas Turbine Engine Pneumatic Power Architecture

The pneumatic power function in the gas turbine engine architecture is accomplished simply by over sizing the engine compressor to meet the pneumatic flow requirements and shaft power requirements. When full airflow is required there is no shaft work demand for hydraulic or electric power. All the air is available for off platform use except the minimum required for proper turbine operation. At low flow rate demands, only enough air to satisfy the off platform requirement is provided and excess air is sent to the turbine or discharged overboard. Small turbine engines typically use a single stage radial flow centrifugal compressor that can achieve mass flow rates at the required pressure ratios of 2.0–3.5 if the rotational speed is high (Heywood 1988). Rotational speeds of radial centrifugal compressors in gas turbine engines can easily exceed 60,000 rpm in GSE applications (Simmons 1968) making them well suited to provide the flows and pressures required. Complexity of the drive pad transmission is reduced by not having to interface a separate piece of equipment for pressurized air.

F. POWER PLANT

Ground support equipment power plants are problematic in Army aviation units. In the era of two-level maintenance, field units have shop facilities for aircraft gas turbine engine repair. In cases where repairs involve the hot section of a gas turbine engine, the repaired item requires testing on a dynamometer under load before being put back into service. There are a dozen deployable aircraft engine test systems known as Flexible Engine Diagnostic System (FEDS). Army aviation uses these systems to perform engine

power tests on-site. This avoids having to send the engine to the aviation depot or approved private sector concern for dynamometer testing. Unfortunately, the FEDS does not support small gas turbine engines, limiting repairs that can be performed by the maintenance unit. Field repairs that involve the diesel engines cannot be performed in aviation units and must be performed by ground units at the motor pool. To further complicate the matter, ground and aviation units cannot order parts for equipment outside their designation, making an efficient logistics strategy extremely important.

Power plant reliability needs to be as high as practicable because any unscheduled maintenance, except a few gas turbine engine auxiliaries, requires the system to be down waiting on another organization to perform the maintenance and return the equipment. Without high reliability crews may opt not to maintain the GSE or remove and replace the power plant for minor issues resulting in negative consequences to tactical operational readiness, excessive cost, or both. Reliability, efficiency, mass, and hazardous waste generation are the primary cost attributes of the power plant architectures.

1. Diesel Engine

The diesel engine is a four-stroke cycle internal combustion engine that relies on the cylinder pressure to ignite the fuel. Combustion occurs at constant pressure and theoretical thermodynamic efficiencies exceed 60% (Severns and Degler 1948). Advances in design, control, and materials have increased practical thermal efficiencies from the 35–40% of 50 years ago to over 50% currently (Perkins Engines Company Limited 2007). Efficiency gains are achieved primarily by using higher compression ratios which have a negative effect on system mass because higher pressures necessitate more material to compensate for the higher induced stresses. The use of turbochargers and aftercoolers offset some of the mass required to generate horsepower but add complexity to an already complex machine. Fortunately, a multitude of companies throughout the world produce these engines on a large scale. This allows good reliability and reasonable cost per unit power.

a. Reliability

Mean time between failures (MTBF) of diesel generator sets is reported to be between 7,000–14,000 hours (System Reliability Center 2001). Component data shows that component MBTF are much higher than the complete generator set; therefore, the assumption is made that the generator set MBTF is approximately that of the power plant. The system should have a service life between depot level refurbishment between 2500 and 5000 hours. Based on an exponential distribution, the diesel power plant reliability is between 86–96%. Other sources report reliability rates for diesel engines used for prime electric generators to be between 96–98% with availability between 90–95% (Boyce 2002). For comparison herein, a reliability estimate of 96% will be used for the diesel engine power plant.

b. Efficiency

Fuel efficiency can best be quantified by relating the fuel consumed to produce shaft horsepower, termed thermal or thermodynamic efficiency. Support equipment rarely operates at the rated load, so how efficiency varies with load is of particular importance. In the early 1970s diesel engine technology could deliver specific fuel consumptions (SFC) in the range of 0.337–0.383 lb_m/hp-hr that translates into thermal efficiencies of 36% to 41% (Schnell 1971). Today reported values of efficiency range from 38–50% (0.277–0.366 lb_m/hp-hr @1800 rpm), with aftercooled turbocharged engines showing the best thermal efficiencies.

A survey of manufacturers' literature shows significant improvements in specific fuel consumption. Data presented in Figure 6 reflect various manufacturers and displacements (101–395 in^3) operating at a shaft frequency of 1800 rpm. The idle load was considered to be 5 hp which is consistent with dynamometer testing of the current gas turbine engine when under a no load condition other than turning the drive pad gearbox (U.S. Army Aviation and Missile Command 2012). The only fuel consumption information found at idle was from a vehicle with an engine idle speed of 1100 rpm. The data point was not used in the curve fit of the data because of the engine speed difference

and is solely presented as a reference to garner confidence in the presented relationship between fuel consumption and shaft power.

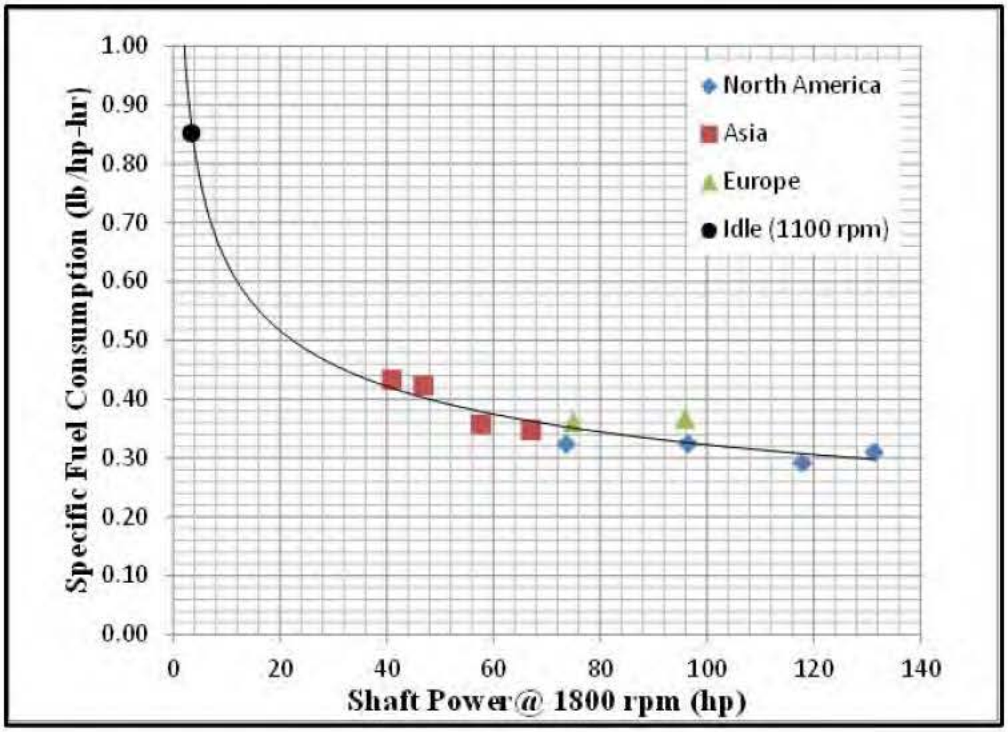


Figure 6. Specific Fuel Consumption Versus Shaft Power

The specific fuel consumption requirements for generator sets producing total power similar to the GSE total power requirement are between 0.31–0.33 lb_m/hp-hr per MIL-DTL-32496 (Department of Defense 2014). Based on Figure 6, the diesel power plant will be assumed to have a specific fuel consumption of 0.37 lb_m-fuel/hp-hr corresponding to 62 hp at 1800 rpm.

c. *Mass*

Diesel engine mass to power ratios are a challenge for mobile power systems. Naturally aspirated engines are the least efficient with respect to fuel consumption, which translates into poor mass to power ratios as well. By turbocharging the engine and adding aftercooling the power is substantially increased with little additional mass (Heywood 1988) as shown in Figure 7.

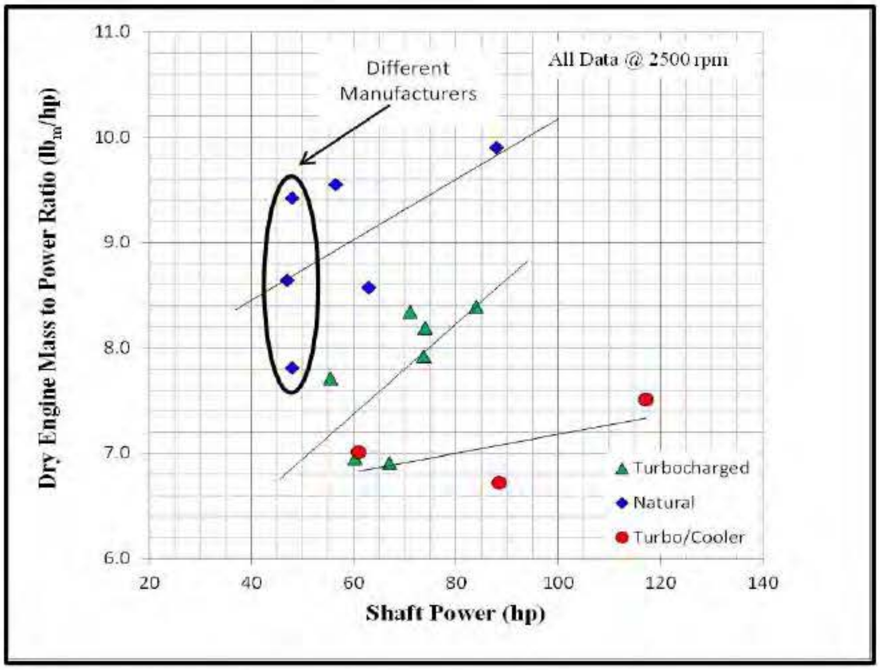


Figure 7. Power to Mass Ratio for Diesel Engines

Data show turbocharging improves power to mass ratio by 12% to 23% at a constant frequency of 2500 rpm. These correlations are not the best and the relationship between shaft power and dry engine weight is not linear. Variations in displacement, compression ratios, and manufacturing further scatter the data. The "Naturally Aspirated" data show significant spread in dry mass to shaft power ratio among three manufactures with similar shaft power engines. Plotting *dy/dx* against *x* helped linearize the data. Though the correlation is poor ($R^2 \approx 0.5$), the trend is correct and is only for use in area where data exist.

The masses used in Figure 7 are reported dry weight of the engine only. The actual power plant deployed in the systems requires control, cooling, mounting structure and other auxiliary equipment for proper installation. Manufacturers offer industrial power units prepackaged to interface with other machines. Since the mass of the engine also affects the size of the cooling system, filtration system, and support structure the prepackaged units were examined to ascertain the relationship between engine dry weight and an integrated power plant weight.

Figure 8 contains these results. When installed in the ground support equipment the power plant components will not be in the same configuration but will include all the components or facsimiles with the same function. Data were only found from one manufacturer that allowed for direct comparison between the dry engine and the industrial power units, so it is assumed that the ratios are approximate for all the manufacturers.

Another important parameter required to perform the analysis of alternatives is the envelope or size required for the power plant. The envelope density for integrated power plants examined was determined to be 31± 3.7 lb_m/ft^3 with the height to length ratio being 0.891±0.110. Knowing how the width of the integrated power plant varies with dry engine weight Figure 9 allows estimates of mass and dimensioned envelope (LxWxH) based solely on the shaft power requirements of the system.

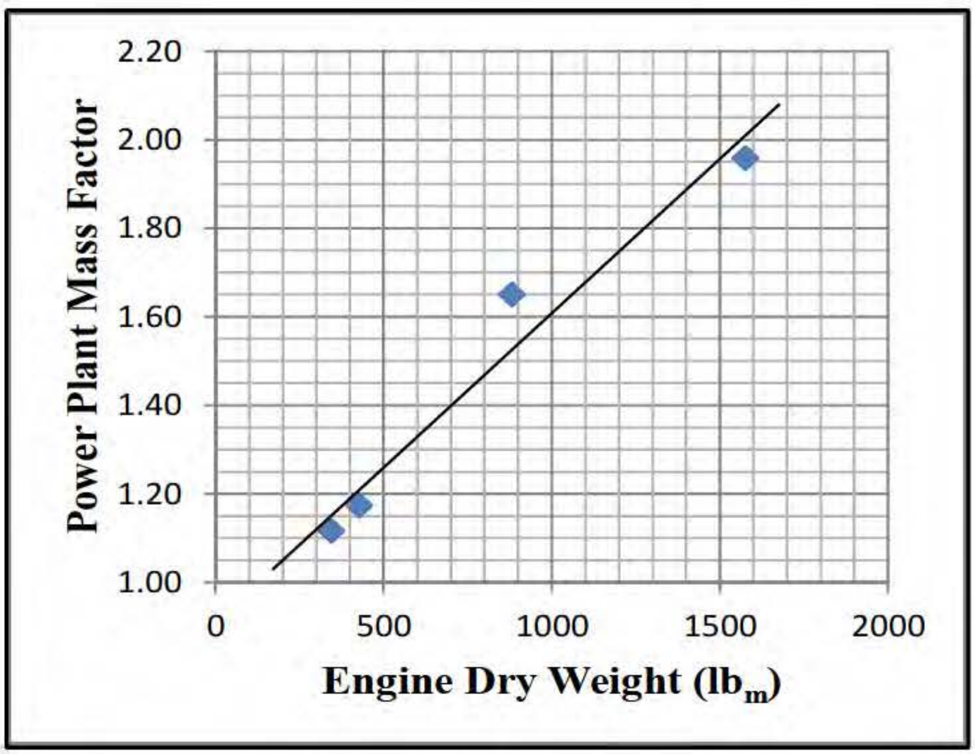


Figure 8. Power Plant Mass Factor

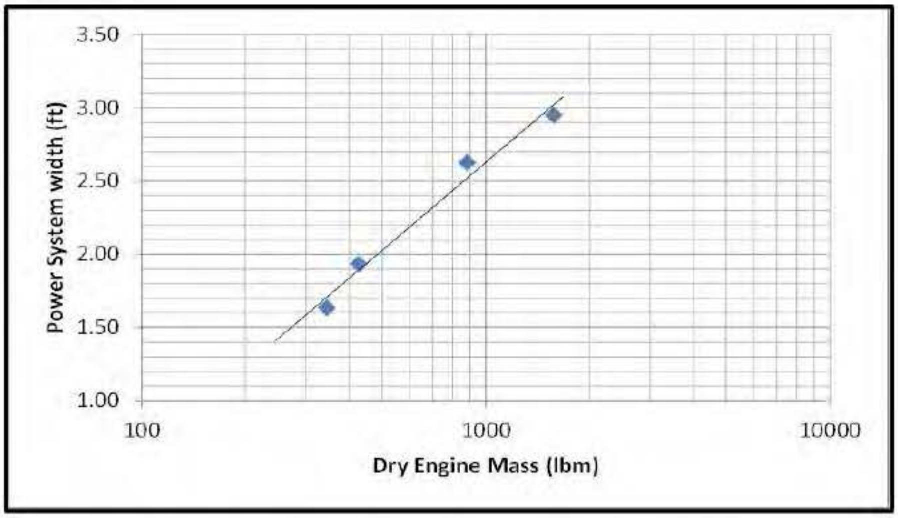


Figure 9. Power Plant Width Versus Dry Engine Mass

A sample calculation for a 100 hp shaft power turbocharged engine yielded a mass of 1341 lb_m and an envelope of length 49–56 inches, width 30 inches, and height of 44–50 inches. Actual reported values from the vendor catalog for a turbocharged industrial power unit are 1455 lb_m with an envelope of 57.1 x 31.5 x 45.3 inches. The discrepancy is reasonable; the catalog item has a reported engine frequency of 2200 rpm which is less than the 2500 rpm on which the predictive data that is based. Higher shaft frequencies typically produce more power for the same displacement, thus a smaller engine may match the power output of larger engines operating at slower speeds. The expectation is that the mass prediction for this item will be low due to the engine speed difference. Development of additional frequency curves or an additional factor to account for difference in shaft frequency can help the mass prediction accuracy but is not warranted for the high level analysis being performed.

d. Hazardous Waste Generation

The generation of waste by the engine is a function of lubricants and cooling system maintenance. Typical diesel engine maintenance usually requires the oil to be changed every 500 hours or six months and the coolant system be serviced annually. This may vary somewhat among manufacturers but not widely. Maintenance is a large cost driver in the private sector and these intervals are becoming prevalent among suppliers.

Given the upper usage bound of 1000 annual operating hours annually provides the basis for waste generation. Engines with the power characteristics in the range of interest to this application will carry between 12–16 quarts of oil and have cooling systems that hold approximately five gallons of coolant. Large oil capacities are necessary to ensure enough oil additives remain in the system to control acid formation, wear, and foaming among other functions. To add some perspective, 500 hours of operation is similar to the average commuter driving 17,500 miles (assumes average speed of 35 mph).

These usage rates will generate an oil waste stream of approximately 60 lb_m/yr and a coolant waste stream of approximately 40 lb_m/yr. Combining these streams yields a value of 0.10 lb_m per hour of operation per end item.

e. Diesel Engine Summary

The maximum engine mass is derived from the maximum allowable system mass of 4500 lb_m less 300 lb_m for diesel engine peculiar items (addition of pneumatic air and structure associated with heavier components), less the current system mass of 3450 lb_m (ship mass less power plant dry weight, exhaust, and mounting hardware). This yields an upper limit engine power system mass target of 750 lb_m.

Table 3 Gives the estimated masses for each diesel engine technology in the shaft power range of interest. Values of 78 and 94 hp are 125% of the maximum continuous shaft power for loads of 62 and 75 hp, respectively.

Table 3. Diesel Power Plant Mass and Envelope Estimate

Power (hp)	Diesel Engine Technology			Envelope Estimate		
	Naturally Aspirated	Turbocharged	Turbocharged Aftercooled	L (in)	W (in)	H(in)
62	733	568	508	39	24	35
75	861	676	571	41	25	37
78	1061	862	686	45	28	40
94	1239	1007	803	49	29	41

The estimates indicate that none of the technologies can meet the mass target at a rated shaft power of 91 hp. Reducing the continuous shaft power requirement to 62 hp lowers the design shaft power to 78 hp.

Using 78 hp as the design point is reasonable as the baseline system has proved suitable in the tactical environment at this performance level. The data in Table 3 are based on shaft frequencies of 2500 rpm. Optimal shaft frequency to operate this type of equipment is around the speed that produces maximum torque, usually around 1800 rpm. This means that there may be additional risk associated with the mass requirement as the detailed design explores the power/torque trade space. Since engines have discrete displacements, smaller displacement engines that may be in the desired power range would be working closer to their maximum power output to meet the required load. This limits the available power/torque trade space for smaller displacement engines perhaps forcing selection of the next larger displacement and the commensurate mass penalty. The analysis indicates the use of a diesel power plant is viable, but the total system mass should be considered a risk with respect to the maximum mass limit of 4500 lb_m.

The power plant for this application is estimated to have the following characteristics: a specific fuel consumption rate of 0.37 lb_m-fuel/hp-hr at 62 hp; a mass of 686 lb_m (excluding pneumatic power function); an envelope of 45 x 28 x 40 (LxWxH) inches (packing density of 19.58 lb_m/ft^3); a reliability of 93%; and a hazardous waste generation rate of 0.10 lb_m per hour of operation.

2. Gas Turbine Engine

The gas turbine engine is a Brayton cycle machine consisting of a compressor, a combustor, and a turbine. Air is compressed, heated at constant pressure by the injection and combustion of fuel, and shaft work is extracted from the system as the air and combustion products are expanded across the turbine. Theoretical thermodynamic efficiencies exceeding 60% can be achieved at pressure ratios above 35 (Boyce 2002). The challenge for this application is that simple single stage systems can only achieve pressure ratios between about 3 and 6 (Simmons 1968)—at these low pressure ratios the theoretical efficiencies are a mere 25%.

The primary benefits of the turbine engine are low mass to power ratio, limited number of parts, and machine simplicity. The least favorable attribute is thermal efficiency, which is a function of the pressure ratio and the turbine inlet temperature. Pressure ratios increase with shaft frequency, compressor stages, or both. Unfortunately, higher turbine inlet temperatures are only effective in increasing efficiency at higher pressure ratios. For single stage systems, increasing the turbine inlet temperature does not appreciably affect system efficiency (Boyce 2002). Operating systems with multiple stages and at higher turbine inlet temperatures have a negative effect on system cost and mass.

Efficiencies can also be gained by cooling air between compressor stages (intercooling) and heating inlet combustion air with exhaust gases prior to entering the combustion chamber. The latter is termed recuperation or regeneration; regeneration is the only option available for single stage systems. These options also have a negative impact on both procurement and maintenance cost, in addition to system mass.

a. Reliability

MTBF of gas turbine generator sets is reported to be between 5,000- 30,000 hours (System Reliability Center 2001). The system must have a depot level inspection every 500 hours. Expected service life between depot level refurbishment is 5000 hours. Based on an exponential distribution, the power plant reliability ranges between 37–85%. Reliability rates for gas turbine engines used for prime electric generators are reported to range between 95–97% with availability between 85–90% (Boyce 2002). For comparison purposes, a reliability estimate of 90% will be used.

b. Efficiency

Gas turbine engines are designed to operate at full load, which is the point of maximum efficiency; the efficiency decreases as the total power output decreases (Simmons, 1968). When operated at low pressure ratios (3.0–5.0), a constant shaft frequency, and with pneumatic power being produced by the engine compressor section, the actual thermal efficiencies range between 8–16% (Department of the Army 2010;

U.S. Army Aviation and Missile Command 2012; Simmons 1968). This translates to a SFC range based on total power of 0.892 to 1.774 lb_m/hp-hr as shown in Table 4.

Table 4. AGPU GTE Performance Data

Pneumatic Power (hp)	Shaft Power (hp)	Total Power (hp)	SFC (lb_m/hp-hr)	η (%)
0	62	62	1.774	7.8
57	46	103	1.214	11.4
116	3	119	0.892	15.5

This type of fuel economy warrants an examination of configuration changes that can be made to the GTE power plant architecture. Excluding multiple stage architectures, the only real option to examine is the addition of a regenerator. A regenerator is simply a heat exchanger that preheats gas exiting the compressor prior to the combustor. The current system turbine exhaust temperature controls to 1250°F, much higher than the compressor exit temperature. Another method to increase efficiency is to allow the compressor to operate at multiple shaft frequencies. This allows operation at the most efficient shaft speed for the load. However, it means that the system would not be allowed to operate a precise generator on a common shaft—a free or split shaft turbine would have to be employed to drive the auxiliaries. Figure 10 schematically shows the current bleed air architecture and the corresponding regenerative free turbine system.

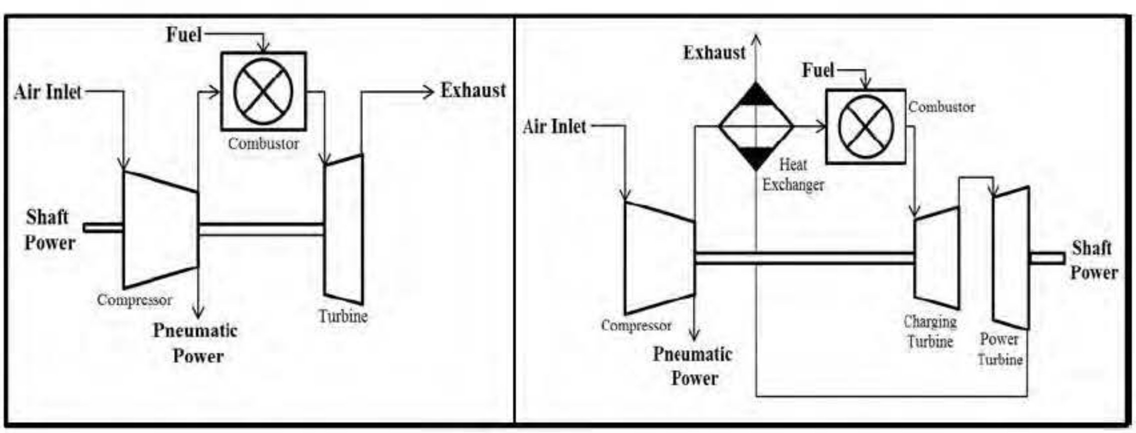


Figure 10. Single Stage Gas Turbine Architectures

The heat exchanger can be added to the current architecture without the addition of the free turbine. At the pressure ratios of a single stage, machine efficiencies of 25–32% (0.432–0.553 lb_m/hp-hr) can be achieved (Schnell 1971) by a free turbine with regeneration.

Some considerations of adding a regenerative heat exchanger are increased cost, envelope, and mass while decreasing reliability and availability due to higher maintenance requirements (Simmons 1968). The need to keep the heat exchanger clean requires increased maintenance. A fouled heat exchanger leads to increased pressure drop and lower heat transfer coefficients which can actually cause the machine to have lower thermal efficiency than a machine with no heat exchanger (Boyce 2002). Figure 11 depicts the mass penalty of efficiency gains by adding heat exchanger surface area at constant power.

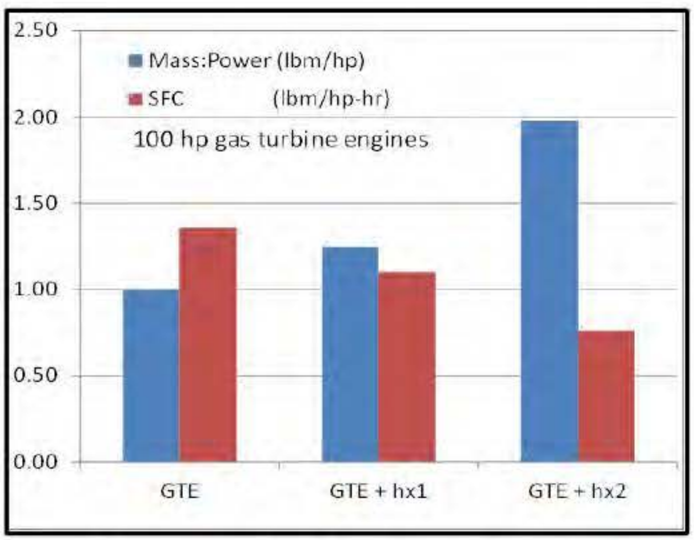


Figure 11. Effect of Regeneration on Mass and SFC (after Simmons 1968)

The free turbine system is able to provide high torque at low shaft frequencies, attractive for the GSE application. Maintaining precise rotational frequency control under varying shaft loads is difficult and requires a sophisticated control system. First, there is a lag between the drive system and the free turbine. Next, there are issues of energy management and control of the free turbine as loads come off line. These conditions are

exacerbated with a regenerative heat exchanger due to the turbine storing more pressure energy created by the additional head required to move air through the heat exchanger (Simmons 1968).

These changes would reduce the primary advantage offered by gas turbine engine technology—low mass to power ratio—and still not be competitive with diesel engine technology on fuel efficiency. The decrease in availability associated with these changes also makes the system less attractive. Additional tasks of cleaning the engine compressor and heat exchanger are not desirable. Therefore, the baseline single stage single shaft system will be the GTE configuration considered most viable for this study.

c. Mass

Simple single shaft gas turbine engines in this power range have mass to power ratios between 0.9 and 1.1 lb_m/hp. This is based on maximum total power. The current engine in the AGPU has a mass of 130 lb_m and has a maximum total power output of 119 hp, yielding a mass to power ratio of 1.09. The envelope for the dry engine is 33x21x25 inches (LxWxH). This is very deceiving because the exhaust system is about as large as the engine itself; the air inlet and filtration system are quite large as well. The system occupies an irregular space. Accommodating the filter, exhaust, and portion of the engine requires a space 45x21x32 inches (LxWxH), with the remaining engine occupying a space of 16x21x25 inches (LxWxH). The entire engine assembly with three 8000 rpm drive pads has a mass of 200 lb_m, an envelope of 22.4 ft^3, and a packing density of 8.94 lb_m/ft^3.

d. Hazardous Waste Generation

The generation of waste by the engine is a function of lubrication maintenance. GTE maintenance requires oil changes every 250 hours or six months (Department of the Army 2010). Given the upper usage expectation of 1000 annual operating hours provides the basis for waste generation computations. The engine has an oil capacity of 2.3 quarts, but only 1.6 quarts located in the sump is drained. Thus, usage rate will generate an oil waste stream of approximately 11.9 lb_m/yr or 0.012 lb_m per hour of operation per machine in use.

e. *Gas Turbine Engine Summary*

The gas turbine power plant for this application is estimated to have a specific fuel consumption rate of 1.774 lb_m-fuel/hp-hr at 62 hp, a mass of 200 lb_m, and a packing factor of 8.94 lb_m/ft^3, a reliability of 90%, and a hazardous waste generation rate of 0.012 lb_m per hour of operation.

G. CHAPTER SUMMARY

The system architectures under examination are the baseline GTE with shaft driven hydraulics, GTE with electric drive hydraulics, diesel with shaft drive hydraulics and pneumatics, and diesel with shaft drive pneumatics with electric drive hydraulics. Each subsystem is evaluated to determine the best technology alternative to provide the function. The attributes of the baseline system architecture are given in Table 5.

Table 5. Baseline System Architecture Attributes

Subsystem	Attributes	Mass (lbm)
Propulsion	Axle/Differintial, 3 hp 28 VDC Electric Motor Drive	baseline
Electric	Prime Electric Power 115/200VAC 3ϕ 400Hz, 48kVA	baseline
Hydraulic	Variable Displacement Pump, 16 gpm, 3300psi discharge, 10psia suction, MIL QUAL	baseline
Pneumatic	GTE Compressor Bleed, 34lbm/min max at 50 psia	baseline
Power Plant	GTE: Single Stage Radial Compressor, Single Stage Centripetal Turbine, 62 shp continuous	baseline
System	Envelope (LxWxH) = 90x58x60 inches Volume = 181.3 ft^3 SFC = 1.774 lbm/hp-hr Reliability = 90% Operator Panel SPL = 96 dBA (TM 1-1730-229-13)	3620

Included in the attribute description are the components that comprise the subsystems. Differences between the estimated change in mass between the proposed architecture and the current system are presented. A summary of the relevant requirements at the system level are also provided: system mass, envelope, fuel consumption, reliability, and noise.

1. GTE: Electric Drive Hydraulics (EDH)

Table 6 presents the attributes of the GTE electric drive hydraulic system architecture.

Table 6. GTE EDH Architecture Attributes

Subsystem	Attributes	ΔMass (lbm)
Propulsion	Axle/Differintial, Hydraulic Motor Drive	-230
Electric	Prime Electric Power 115/200VAC 3ϕ 400Hz, 48kVA + inverter	+180
Hydraulic	(QTY 2) 8 gpm COTS Variable Displacement Pumps, suction boost pump, electric motors	+160
Pneumatic	GTE Compressor Bleed, 34lbm/min max at 50 psia	0
Power Plant	GTE: Single Stage Radial Compressor, Single Stage Centripetal Turbine, 62 shp continuous	0
System	Envelope (LxWxH) = 90x58x60 inches Volume = 181.3 ft3 SFC = 1.774 lbm/hp-hr Reliability = 90% Operator Panel SPL = 96 dBA (TM 1-1730-229-13)	3730

An inverter is added to the electric subsystem of this architecture to convert 28 VDC power to 200/115 VAC 3ϕ 400Hz power to drive the electric motors for the hydraulic pump increasing the subsystem mass by 180 lb_m. The hydraulic system utilizes two 8 gpm COTS pumps fed by a reservoir boost pump. Since only one pump is required to drive the hydraulic motors for the propulsion function, the impact on inverter size is minimized. Using electric driven hydraulics increases the hydraulic subsystem mass primarily due to the electric motors and addition of the additional pump in the reservoir. Utilizing 400 Hz power minimizes the electric motor mass penalty, but the impact is still a rather large subsystem mass increase of 160 lb_m, which includes the mass of the weightier COTS pumps.

This configuration has the benefit of lowering the propulsion system mass to 150 lb_m, resulting in a net savings of 230 lb_m. Although there is an overall increase in mass of 110 lb_m this remains a viable architecture that could be easily packaged in the baseline envelope and is well within the mass constraints.

2. Diesel Engine: Direct Drive Hydraulics (DDH)

Table 7 presents the attributes of the diesel shaft drive hydraulic system architecture. The hydraulic system utilizes two shaft driven 8 gpm COTS pumps supplied by an electrically driven reservoir boost pump resulting in a 70 lb_m increase over the baseline.

Table 7. Diesel DDH Architecture Attributes

Subsystem	Attributes	ΔMass (lbm)
Propulsion	Axle/Differintial, 3 hp 28 VDC Electric Motor Drive	0
Electric	Prime Electric Power 115/200VAC 3ϕ 400Hz, 48kVA	0
Hydraulic	(QTY 2) 8 gpm COTS Variable Displacement Pumps, suction boost pump	+ 70
Pneumatic	Centrifugal & Reciprocating Compressors, 26lbm/min max at 50 psia	+ 210
Power Plant	Diesel: Supercharged/Aftercooled, 62 shp continuous	+ 486
System	Envelope (LxWxH) = 90x60x72 inches Volume = 225 ft3 SFC = 0.361 lbm/hp-hr Reliability = 96% Operator Panel SPL = 85 dBA (MIL-DTL-32496/10)	4386

This configuration utilizes the baseline propulsion subsystem. The pneumatic subsystem is comprised of two compressors, a reciprocating low flow compressor and a centrifugal high flow compressor adding an additional 220 lb_m to the system. The diesel power plant adds an additional 486 lb_m. Even though there is an overall increase in mass of 776 lb_m this remains a viable architecture that could be packaged within the envelope constraints. This architecture has just over 20% mass growth available.

3. Diesel Engine: Electric Drive Hydraulics

Table 8 presents the attributes of the diesel engine electric drive hydraulic system architecture. This architecture adds an inverter to the electric subsystem. The hydraulic system utilizes two 8 gpm COTS pumps with electric motors along with a reservoir boost pump. The diesel engine and the dual compressor pneumatic system prove to be a significant mass penalty for this architecture. Though the mass estimate for the system does not exceed the maximum requirement it provides no mass growth margin. Changes to the requirements to remove the high flow compressed air as a standard feature to a

feature that could be added as a kit could allow this architecture to have a more suitable mass. Even with the high mass the system is projected to be well within the envelope requirements.

Table 8. Diesel EHD Architecture Attributes

Subsystem	Attributes	ΔMass (lbm)
Propulsion	Axle/Differintial, Hydraulic Motor Drive	-230
Electric	Prime Electric Power 115/200VAC 3ϕ 400Hz, 48kVA + inverter	+180
Hydraulic	(QTY 2) 8 gpm COTS Variable Displacement Pumps, suction boost pump, electric motors	+160
Pneumatic	GTE Compressor Bleed, 34lbm/min max at 50 psia	0
Power Plant	GTE: Single Stage Radial Compressor, Single Stage Centripetal Turbine, 62 shp continuous	0
System	Envelope (LxWxH) = 90x58x60 inches Volume = 181.3 ft3 SFC = 1.774 lbm/hp-hr Reliability = 90% Operator Panel SPL = 96 dBA (TM 1-1730-229-13)	3730

4. Conclusion

All the proposed architectures are viable with respect to mass, envelope, and performance with the Diesel/Electric Hydraulic architecture. The only caveat is total mass will be larger than ideal. Improved mass could be possible with changes in material selection or in trading functionality and complexity for mass, if justified by a commensurate lowering of life cycle cost.

IV. COST ANALYSIS

A. INTRODUCTION

The cost analysis focuses on the cost differentials created by the differences in architectures. This approach is a variation of the unit-operations and functional unit estimating technique for fixed capital investment (Peters and Timmerhaus 1980). In addition, the variation of life cycle cost will be captured by procurement (fixed capital) as well as operation and support costs. GSE items do not require a research and development phase, these items are COTS or non-developmental items (NDI). Since there is little variation in materials of construction among the candidate systems, differences in disposal costs are negligible. All cost values have been computed in calendar year 2014 US$ unless otherwise stated.

Cost estimating relationships (CER) are used when available and then adjusted to current year dollars using the appropriate producer price index industry data from the Bureau of Labor Statistics. When CERs are not available, a market survey was conducted to obtain a price estimate. With insight into the major components and an analogous system this technique can be expected to yield cost estimates within ± 20% (Holland, Watson and Wilkinson 1984).

B. PROCUREMENT

1. Propulsion System

The equipment costs of the two methods proposed to propel the GSE are similar, except that the hydraulic propulsion requires more sophisticated control. Instrumentation and control cost can vary between 15% and 93% of purchased equipment costs (Peters and Timmerhaus 1980). Since the actual equipment being installed is COTS, a specialized controller will be similar to the cost of the propulsion system equipment. The controller cost is estimated to be 93% of the $5000 hardware cost or $4,650 which represents the cost difference between the two propulsion methods.

2. Electric System

The only difference in the electric systems proposed is the addition of the inverter to power the hydraulic system from the battery pack for architectures that have hydraulic powered propulsion. The price of an NDI military qualified inverter to convert 24 VDC power from the battery pack to 200Y/115VAC 3ϕ 400Hz power is $32,000.

3. Hydraulic System

There are three configurations of hydraulics systems: the baseline; two direct drive COTS pumps with a boost pump in the reservoir; and two electric drive COTS pumps with a boost pump in the reservoir. The baseline hydraulic pump is $21,000/unit. The COTS pumps are $8,050/unit with the boost pump being $500. The electric motors to drive the pumps are $6000/unit and the reduction gear drives are $1500/unit. Hydraulic system procurement costs relative to the baseline are as follows:

(a) Direct Drive COTS Pumps: $16,600 – $21,000 = $(4,400)

(b) Electric Drive COTS Pumps: $31,600 – $21,000 = $10,600

4. Pneumatic System

Reciprocating and centrifugal compressors converge in price per pneumatic power supplied as the devices become smaller (Peters and Timmerhaus 1980). A survey of bare oil-less dual stage reciprocating compressors ranging from 5hp to 30hp from four manufacturers gave an average price of $210/hp with a standard deviation of $34/hp. The system requires 47hp of total capacity, $9870, and a gear-box for the centrifugal compressor, $1500, for a total cost of $11,370 for the direct drive configuration. An electric drive configuration of the high flow compressor requires an additional $6,000 for the electric motor yielding a total cost of $17,370.

5. Power Plant

a. Diesel Engine

The current year cost for a diesel engine power plant ready for industrial use is reported to be $160/bhp (Fraizer 2014). Using 78 hp, the power rating required to support

a continuous load of 62 hp, yields an estimated diesel power plant cost of $12,320. This excludes any pneumatic power functionality.

b. Gas Turbine Engine

The estimated 2008 USD cost for a continuous 62 hp aeroderivitive gas turbine engine power plant was $1330/hp (Pauschert 2009). Using the "turbines and turbine generator sets" producer price index industry data, the current year price estimate for a continuous 62 hp gas turbine engine is $90,290. This does include all pneumatic power functionality.

6. Procurement Cost Summary

Table 9 illustrates the differential procurement costs. Gas Turbine engine costs are the dominant cost driver for the unit and significant savings can be realized by using a diesel power plant. Using electric powered hydraulics proves to be a significant additional fixed capital expense that must be justified by operational efficiency gains of aviation maintenance units and readiness. Negative costs are in terms of required outlays against the baseline; therefore the lowest values are preferred.

Table 9. Differential Procurement Cost Summary

Architecture	System					Total Δ
	Propulsion	Electric	Hydraulic	Pneumatic	Power Plant	Unit Cost
Baseline	N/A	N/A	N/A	N/A	N/A	0
GTE EDH	$4,650	$32,000	$10,600	$0	$0	$47,250
Diesel DDH	$0	$0	($4,400)	$11,370	($77,970)	($71,000)
Diesel EDH	$4,650	$32,000	$10,600	$17,370	($77,970)	($13,350)

C. OPERATION AND SUPPORT (O&S)

Of the six elements of O&S costs the changes in hardware discussed herein only affect unit operation and maintenance costs. Costs of interest from these elements are operating material (fuel, filters, and lubricants), maintenance repair materiel (spare parts)

(Cost Analysis Improvement Group 2007), and item overhaul (Gille 1978). All other O&S cost elements are the same between the presented architectures.

1. Propulsion System

The additional maintenance cost for the hydraulic drive can be estimated by taking 6% of the increased fixed capital cost; this yields $280/yr (Peters and Timmerhaus 1980).

2. Electric System

The additional maintenance cost for the power inverter can be estimated by taking 8% of the increase in fixed capital cost; this gives $2560/yr (Peters and Timmerhaus 1980).

3. Hydraulic System

There are three configurations of hydraulics systems—the baseline, two direct drive COTS pumps with a boost pump in the reservoir, and two electric drive COTS pumps with a boost pump in the reservoir. The annual maintenance costs for variable displacement hydraulic pumps are approximately 15% of the initial fixed capital investment (Peters and Timmerhaus 1980). The electric motors to drive the pumps are $6000/unit and the reduction gear drives are $1500/unit. The motors have a maintenance factor of 2% of initial cost (Fraizer 2014) while the reduction gears will be calculated at 6% of equipment cost. Hydraulic system maintenance costs are as follows:

(a) Baseline: $3150/yr

(b) Direct Drive COTS Pumps:

$8,050/pump x 0.15 x 2 pumps = $2415/yr

$500/pump x 0.06 x 1 pump = $30/yr

TOTAL $2445/yr

(c) Electric Drive COTS Pumps:

$8,050/pump x 0.15 x 2 pumps = $2415/yr

$500/pump x 0.06 x 1 pump = $30/yr

$6,000/motor x 0.02 x 2 motors = $240/yr

$1500/drive x 0.06 x 2 drives = $180/yr

TOTAL $2865/yr

4. Pneumatic System

There are two potential alternate pneumatic power configurations. The alternate configurations have one reciprocating (7.5hp) and one centrifugal compressor (40hp). The use rate between these items is quite divergent. The small compressor will be used regularly while the large compressor will see little if any use. To account for this the small compressor maintenance cost will be estimated at 15% of the original equipment price and the large compressor will be estimated at 2% of the original equipment price. For the high flow electric drive, 2% will be used for all components.

(a) Direct Drive Compressors:

$1575/small compressor x 0.15 = $236/yr

$8400/large compressor x 0.02 = $168/yr

$1500/gearbox x 0.02 = $ 30/yr

TOTAL $434/yr

(b) Electric Drive High Flow Compressor:

$1575/small compressor x 0.15 = $236/yr

$8400/large compressor x 0.02 = $168/yr

$6000/electric motor x 0.02 = $120/yr

$1500/gearbox x 0.02 = $ 30/yr

TOTAL $554/yr

5. Power Plant

a. Diesel Engine

Determining operation and maintenance cost for a power plant is a difficult proposition. Military equipment is not operated nor maintained the same as commercial equipment. Information in the literature about commercial diesel generator systems suggest a repair and maintenance factor (RMF) as a percentage of equipment costs but little guidance is provided on how the factors change with critical parameters. To

compensate for this, historical army O&S cost data taken from research performed by Gille (1978) are used to gain insight to how cost parameters differ in a military organization versus commercial practice and establish a reference point for estimating current year costs. This analysis uses four cost categories to capture operation and maintenance cost estimates for the diesel engine power plant: fuel consumption, repair parts (spares and consumables), item overhaul, and hazardous waste disposal.

Fuel costs for the diesel power plant are a function of the fuel price and the engine efficiency. The specific fuel consumption derived from the study by Gille (1978) was determined to be 0.490 lb_m-fuel/hp-hr for full rated engine shaft power output ranging from 20–300 hp. This value does not reflect efficiency gains in engine technology, nor does it reflect the relationship between output power and fuel consumption. The fuel cost estimate is based on 1000 operating hours, 50% at full load and 50% at half load which produces an effective operational SFC of 0.40 lb_m-fuel/hp-hr per Figure 6. Using this SCF, an average shaft load of 46.5 hp, the Defense Logistics Agency (DLA) FY2014 standard energy price of $3.62/gal for JP-8 fuel, and a fuel API gravity of 44 (Department of Defense 2013) yields an annual fuel cost of $10,016/yr ($0.2154/hp-hr) per unit.

Repair part costs include filters, lubricants, soft goods, and spares excluding materiel required for overhaul. To obtain an estimate, costs were escalated from the Gille (1978) study which used four categories to capture the O&S costs: fuel, spare parts, overhaul, and labor. The idea is to use known current year (2014) and then year (1978) costs to create a relationship between the Gille (1978) cost elements that are applicable to the current year. Of the cost elements, fuel and labor are known in current year dollars. Fuel costs are discussed previously. The labor costs are escalated using the military labor rates for the applicable years with the assumption that the labor expended per hour of equipment operation is comparable. With two of the four cost elements known in current year dollars and no reliable method of independently escalating the spares or overhaul costs, a correlation relating the current year known elements and the current year unknown elements is required. The 1978 O&S cost data is used to determine that relationship.

To develop that relationship, the Gille (1978) spares and overhauls costs are combined into one cost element to reduce the number of variables. Developing an escalation relationship based on fuel costs is not reliable because fuel costs have not escalated at the same pace as labor and spare parts. Therefore, a function relating labor costs (LC) and the new construct of spare parts and overhaul (PARTS) costs is used. The function providing the best relationship is the ratio of PARTS and LC as a function of LC divided by the total O&S costs. This relationship showed a reasonable correlation (R^2=0.85) and is presented in Figure 12.

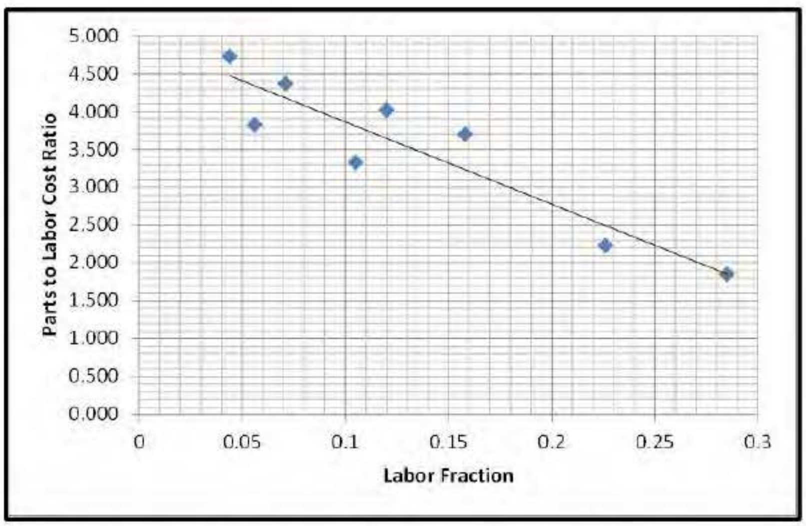


Figure 12. Parts and Labor Costs Correlation

Using the current year fuel and labor costs, a cost for PARTS is guessed and subjected to two constraints—the relationship depicted in Figure 12 and sum of the cost fractions equal one. The estimate for PARTS in current year dollars is the cost that satisfies these constraints. The individual current year costs for spares and overhaul are then calculated for each shaft power using the ratio of the spare parts and overhaul cost reported by Gille (1978). Table 10 presents the Gille (1978) then year costs and the estimated current year costs for shaft powers between 7.5 and 300 hp.

Table 10. Engine Power Output and Related Operating Cost (from Gille 1978)

full load shaft power (hp)	Operation & Maintenance Cost (US$/hp-hr)									
	Fuel		Parts		Overhaul		Labor		Total	
	CY$1978	CY$2014	CY$1978	CY$2014	CY$1978	CY$2014	CY$1978	CY$2014	CY$1978	CY$2014
7.5	0.0627	0.4134	0.0711	0.2703	0.1048	0.3984	0.0951	0.2100	0.3337	1.2921
15	0.0543	0.3377	0.0404	0.1757	0.0603	0.2619	0.0453	0.1285	0.2002	0.9038
22	0.0487	0.3000	0.0438	0.1823	0.0664	0.2766	0.0298	0.1416	0.1887	0.9005
45	0.0476	0.2450	0.0223	0.1059	0.0354	0.1686	0.0144	0.0776	0.1197	0.5970
90	0.0464	0.2001	0.0121	0.0623	0.0176	0.0908	0.0089	0.0398	0.0850	0.3929
150	0.0457	0.1724	0.0084	0.0473	0.0145	0.0813	0.0052	0.0332	0.0738	0.3341
225	0.0446	0.1531	0.0052	0.0306	0.0079	0.0465	0.0034	0.0187	0.0611	0.2489
300	0.0028	0.1408	0.0040	0.0259	0.0085	0.0558	0.0026	0.0202	0.0179	0.2428

Based on Table 10, the annual parts cost for a 62hp engine operating 1000 hours per year is $4,803 ($0.0775/hp-hr) per item. The annual overhaul costs are estimated to be $7,738 ($0.1248/hp-hr). Note that since costs/energy ($/hp-hr) for these data are based on the full load shaft power for the item, erroneous estimates would be obtained if the average energy output of the device were used.

The final component of the operations and support cost is the waste disposal cost. The cost per pound of waste oils and lubricants disposal is $0.933/lb$_m$ (Kim et al., 1991) in CY2014 dollars. The estimated waste generation rate of 0.10 lb$_m$/hr yields an annual cost of $94/yr ($0.094/hr) per item. This term is specific for engines in the range of interest to this study, specifically 50–100hp. A cost per energy can be obtained, applicable to this study only, by dividing the cost per hour by the full load shaft power yielding $0.0015/hp-hr.

The annual total diesel engine operating cost in CY2014 dollars per item excluding overhaul costs is estimated to be $14,913 and $22,651 including overhauls. A cost estimate based on diesel engine use in commercial applications (power generation and irrigation) for parts, fuel, filters, and lubricants yields estimated costs between $12,183 and $13,080 (Fraizer 2014; United States Agency International Development 2011; Peters and Timmerhaus 1980). These values are slightly lower that the estimate derived from the escalation of the Gille (1978) data; this is not unexpected for several reasons. First, commercial applications often use equipment at engine power levels commensurate only with the larger units in the Gille (1978) study. Secondly, the equipment is operated at the maximum rated load for the application. Both of these

factors tend to be more efficient than typical military equipment operation. Contract cost also flow into the parts and commodities that effect government procurement costs that are not present in commercial transactions. Thus the estimates appear reasonable and are consistent with military cost categories.

b. Gas Turbine Engine

As with the diesel power plant, four areas will be considered in the operating and maintenance cost of the gas turbine engine power plant: fuel consumption, repair parts (spares and consumables), item overhaul, and hazardous waste disposal.

Fuel costs for the gas turbine power plant are a function of the fuel price and the engine efficiency. Fuel consumption cost determination is based on 1000 operating hours, 50% at full load and 50% at half load. The composite SFC from Table 4 at the average shaft load of 46 hp is 1.214 lb_m-fuel/hp-hr, yielding an annual fuel cost of $30,072/yr ($0.6537/hp-hr) per unit.

The repair parts costs include filters, lubricants, soft goods, and repair parts excluding overhaul. An estimate for this cost category comes from fixed and variable cost from commercial power systems. Gas turbine engines have a much lower costly maintenance burden compared to diesel engines. Data indicate that in systems with equivalent power output the gas turbine engine total operation and maintenance cost is 55.3% of its diesel counterpart (Boyce 2002). The commercial operation and maintenance does not match well with the cost categories presented for the diesel engine. These data certainly include labor, overhead, taxes, insurance, and depreciation. The data also indicate that the systems have the same thermal efficiencies. This implies the gas turbine data is derived from a much larger and more complex machine than the gas turbine being considered in this study. The benefit of having the same thermal efficiency is that a ratio of the costs eliminates the fuel cost and should also negate the other undesired cost elements. Using 80% of this factor to adjust for complexity yields an annual parts cost of $2,125 ($0.0343/hp-hr based on continuous rated shaft power).

The overhaul cost for the baseline gas turbine engine is about 35% of the equipment cost based on proposal market research. These systems, as discussed earlier,

are sent to the depot every five years; therefore, the overhaul cost is obtained by taking 35% of the equipment cost and spreading that cost over the time period. This gives an annual overhaul cost of \$6,320/yr (\$0.1019/hp-hr).

The waste disposal cost for the estimated waste generation rate of 0.012 lb_m/hr yields an annual cost of \$11/yr (\$0.011/hr) per item. To calculate a cost per energy, divide the cost per hour by the full load shaft power to yield \$0.0002/hp-hr.

The annual total GTE operating cost in CY2014 dollars per item excluding overhaul costs is estimated to be \$32,536 and \$38,856 including overhauls.

6. Operations and Support Cost Summary

The operations and support cost differentials with respect to the baseline are presented in Table 11. Fuel costs are the dominant cost driver in this cost category. Diesel engine parts demand and the inverter for the hydraulic motor drives are secondary and tertiary. Hazardous waste disposal costs are combined with fuel costs.

Table 11. Differential O&S Cost Summary

Architecture	System	Cost Element ($CY2014/yr-unit)				Total Δ
		Parts	Overhaul	Fuel/Waste	Sub Total	Unit Cost
Baseline	All	N/A				0
GTE EDH	Propulsion	280			280	\$2,555
	Electric	2,560			2,560	
	Hydraulic	(285)			(285)	
	Pneumatic					
	Power Plant					
Diesel DDH	Propulsion					(\$16,600)
	Electric					
	Hydraulic	(705)			(705)	
	Pneumatic	434			434	
	Power Plant	2,125	1,518	(19,972)	(16,329)	
Diesel EDH	Propulsion	280			280	(\$13,340)
	Electric	2,560			2,560	
	Hydraulic	(285)			(285)	
	Pneumatic	434			434	
	Power Plant	2,125	1,518	(19,972)	(16,329)	

E. TOTAL LIFE CYCLE COST

The point estimate differential total life cycle cost (LCC) is calculated for each architecture based on a program consisting of 720 units procured and placed in service over a five-year period. Part costs are applied to units the year after being put into service. Overhaul costs are assessed using a five-year gradient, with no cost being assessed in year two and full cost being assessed in year six and beyond. A 20-year unit life was assumed. From years 20 to 25 of the program, a gradient is also applied to the overhaul cost while parts costs continue to be assessed until the final year and are rapidly reduced to zero. Office of Management and Budget (OMB) discount rates (Burwell 2014) are used for the cost analysis. A summary of the results is presented in Table 12, with detailed data presented in Appendix A.

Table 12. Differential Present Values for Various Architectures

Architecture	Differential Costs from Baseline (US$CY2014)	
	Program Present Value	Unit Present Value
Baseline	$0	$0
GTE EDH	$58,094,212	$8,686
Diesel DDH	($211,904,370)	($294,312)
Diesel EDH	($171,488,342)	($238,178)

The architectures featuring the diesel engine show the potential for significant saving in LCC primarily due to fuel efficiency with a present value cost savings of over $294,000 per unit over the 20-year service life. The secondary cost benefit to the diesel engine is the fixed capital cost. The power plant with pneumatic capability has a present worth savings of $64,800/unit over the gas turbine baseline. From a present value point estimate cost perspective the gas turbine architectures are not competitive due to initial GTE cost and specific fuel consumption.

F. CHAPTER SUMMARY

This chapter describes program and unit differential life cycle cost point estimates developed for each of the architectures. Each proposed change in the baseline architecture is assessed for influence in both procurement and O&S costs. No labor

charges are considered. Cost profiles for operation and support are based on diesel engine generator data from 1978 with all labor removed from the costs. Therefore the results only reflect hardware cost associated with the end items and no potential training cost savings associated with electric hydraulic drives is afforded these systems.

The greatest influence on differential life cycle cost is power plant fuel efficiency followed by power plant procurement cost. Figure 13 shows lines of equivalent present value of the differential life cycle costs between the baseline and the two diesel architectures as a function of differential fixed capital cost of the power plant and the differential specific fuel consumption. The triangular area between the line and the ordinate define the space where the baseline has lower life cycle costs. The square black point locates the point estimate value for diesel engine in terms of initial price and SFC with respect to the baseline.

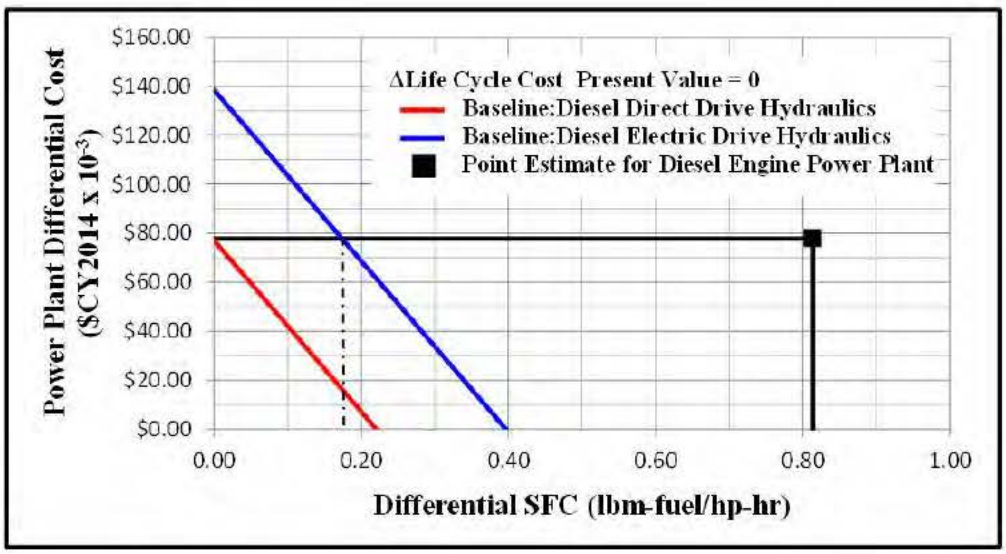


Figure 13. Equivalent ΔLCC for Baseline and Diesel Architectures

The data show that at the current diesel engine price point there is no positive value of differential SFC that will provide an equivalent present value between the baseline and the diesel with direct drive hydraulics. For the diesel with electric drive hydraulics, the data indicate that at the current fixed capital cost differential the baseline becomes cost neutral at a ΔSFC of 0.18 lb_m-fuel/hp-hr. This equates to a SFC of 0.58 lb_m-

fuel/hp-hr versus the value of 1.774 lb_m-fuel/hp-hr used in this study, a reduction of two-thirds, or an efficiency increase in shaft power production to 23.8%. The gas turbine power plant with electric drive hydraulics is completely dominated by the baseline as would be expected with the intangible benefits related to training having to be weighed against the increased cost of the architecture.

Comparing both GTE and diesel with electric drive hydraulics provides the best direct comparison of the two power plants. Figure 14 shows the comparison in differential cost and fuel efficiency as above.

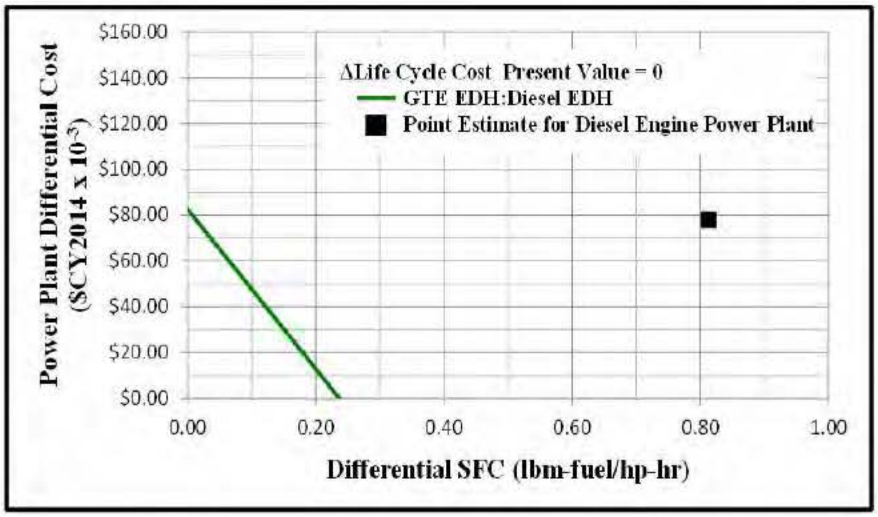


Figure 14. Equivalent ΔLCC GTE and Diesel EDH Architectures

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V. CONCEPT SELECTION

A. INTRODUCTION

Concept selection has to be based on more than point estimations in order to provide confidence in how the life cycle costs for the various architectures compare when the uncertainties surround factors that produce these estimates. Performing a Monte Carlo simulation could accomplish this task. Being a bottom up approach, this method requires probability distributions be created for all the inputs, not a straightforward process based on how many of the parameters are derived. An alternate methodology is the Enhanced Scenario–Based Method (eSBM) which is a top down approach that relies on the projected shape of the cost probability distribution, the point estimate, and an informed judgment of where the point estimate falls on the cost probability distribution (Garvey, Flynn, Braxton, and Lee 2012).

In a traditional eSBM analysis a protect scenario is chosen to characterize a realistic set of risks, that if realized, would impose unfavorable cost consequences. This outcome can then be compared with the program cost probability distribution to determine if the funding levels are adequate. Programs tend to have a lognormal probability distribution in that the costs are more likely to exceed initial estimates rather than achieve cost savings. In this case, cost differences are estimated against a baseline program which may or may not have some lognormal cost probability characteristics. In the examination of the differences between architectures composed solely of COTS and NDI components, the assumption that cost estimates are adequately represented by a normal probability distribution is reasonable. Instead of developing a cost risk scenario, of interest here is the probability of cost equivalency with the baseline and the effect of the distribution variability, termed the coefficient of variation (CV), to establish a cost range for a particular confidence level (Air Force Cost Analysis Agency 2007).

B. ENHANCED SCENARIO–BASED METHOD ANALYSIS

The eSBM technique utilized here treats the differential LCC point estimate as the eSBM program point estimate (PE) and examines the probability of excursions against

that point estimate based on historical variations in CV for similar procurements. These procurements may be for similar programs or elements within programs using equipment similar to that in this study.

1. Parameter Determination

a. Point Estimate (PE)

The point estimates are the values of the differential unit present value costs reported in **ERROR! REFERENCE SOURCE NOT FOUND.**.

b. Point Estimate Probability (α_{PE})

The research conducted by Garvey et al. (2012) does not consider programs on the scale of ground support equipment. GSE is predominantly a sub component of larger weapon system programs. However, in their case study of the NATO Alliance Ground Surveillance system they report that the baseline costs are anchored at the median with program spares and support equipment considered normally distributed. This is also consistent with the notion that the derived costs fall within a symmetrical band about the point estimate. Based on these factors the cost distribution is considered normal and the probability that the program will be less than the point estimate will be assumed to be at the median value, which for a normal distribution is also the mean. Thus for these analyses:

$$\boldsymbol{P}(\Delta \mathrm{LCC}_{\mathrm{unit}} \leq \mathrm{PE}) = \alpha_{\mathrm{PE}} = 0.5 \qquad (1)$$

c. Coefficient of Variation (CV)

The coefficient of variation is a measure of the ratio between the standard deviation and the mean of the probability distribution. Based on the non-developmental

$$CV = {}^{\sigma}\!/_{\mu} \qquad (2)$$

nature of the equipment under evaluation it is most akin to programs at Milestone C. Following guidelines put forth by Garvey et al. (2012), CV values between 0.2 and 0.3 are chosen for the sensitivity analysis for the GTE EDH while the Diesel DDH and Diesel EDH will be given a larger range of 0.2 to 0.5 due to the mass requirement risk

associated with these architectures. These ranges of CV are consistent with criteria Garvey et al. (2012) presented from quantity adjusted then year cost data for Milestone C.

2. Analytical Approach

The eSBM uses the parameters PE, α_{PE}, and CV to calculate a mean and standard deviation for the cost data. For normal distributions the following applies:

$$\mu = PE - z\sigma \tag{3}$$

$$\sigma = \frac{(CV)PE}{1+z(CV)} \tag{4}$$

In the case of normal distributions where α_{PE} is chosen at the median, the standard normal random variable, z, is equal to zero—simplifying the equation (3) to μ = PE and equation (4) to σ = (CV)PE. From these data the probability distribution function of the differential life cycle cost is created allowing evaluation of cost variability of the program at various confidence levels. To be consistent with the 2009 Weapon System Acquisition Reform Act (WSARA) the range of differential costs at the 80% confidence level are examined.

Architectures with differential costs greater than the baseline have probability distribution functions generated using the Excel function Norm.Dist(x,μ,σ,TRUE) over $-5\sigma \leq x \leq 5\sigma$. The distribution communicates the confidence level of the differential cost against the baseline, the confidence level for x=0 represents the likelihood of equivalent present value between the architectures. The 80% confidence level yields the cost using the WSARA criteria. For architectures with greater differential costs compared to the baseline, this sets the reasonable upper bound of the anticipated cost.

Architectures with differential costs less than the baseline have probability distribution functions generated using 1-{Norm.Dist(x,μ,σ,TRUE)} over a range of $-5\sigma \leq x \leq 5\sigma$. As before, the confidence level for x=0 represents the likelihood of equivalent present value between the architectures. The difference is that the 80% confidence level yields the reasonable bound of cost savings and is less than the point estimate.

C. SENSITIVITY ANALYSIS

1. GTE EDH

The probability distribution functions for the differential life cycle cost of the gas turbine engine electric drive hydraulics architecture are shown in Figure 15. The range of maximum expected differential cost with respect to the baseline is between \$10,145 and \$10,875 in CY2014 dollars based on a point estimate of \$8,686. The point estimate plus 20% yields a value of \$10,423.

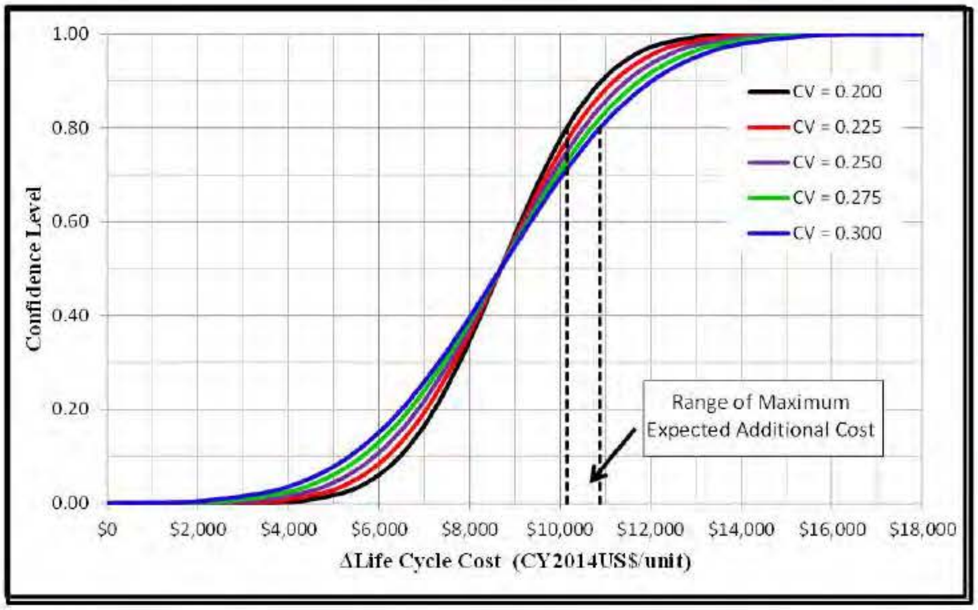

Figure 15. GTE EDH ΔLCC Probability Distribution

2. Diesel DDH

The probability distribution functions for the differential life cycle cost for the diesel engine direct drive hydraulics architecture is shown in Figure 16. The data indicate that present value equivalency occurs at confidence levels between 97.77% and 100%, meaning that this architecture has a probability in this range of being equal to or less than the baseline architecture cost. The range of minimum expected differential savings with respect to the baseline is between \$170,701 and \$244,868 based on a point estimate of \$294,312. The point estimate minus 20% yields a value of \$235,674.

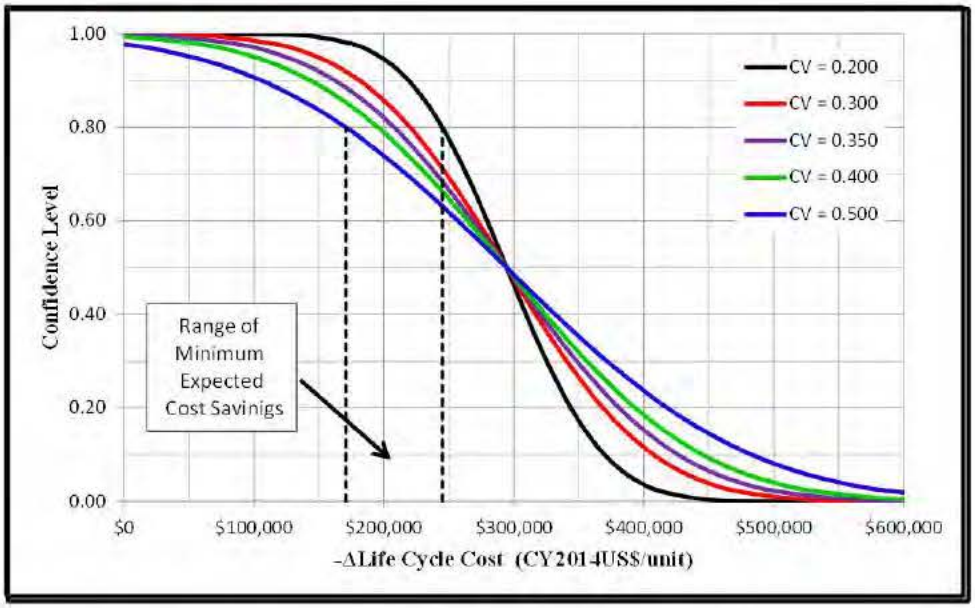

Figure 16. Diesel DDH ΔLCC Probability Distribution

3. Diesel EDH

Figure 17 depicts the probability distribution functions for the differential life cycle cost for the diesel engine electric drive hydraulics architecture.

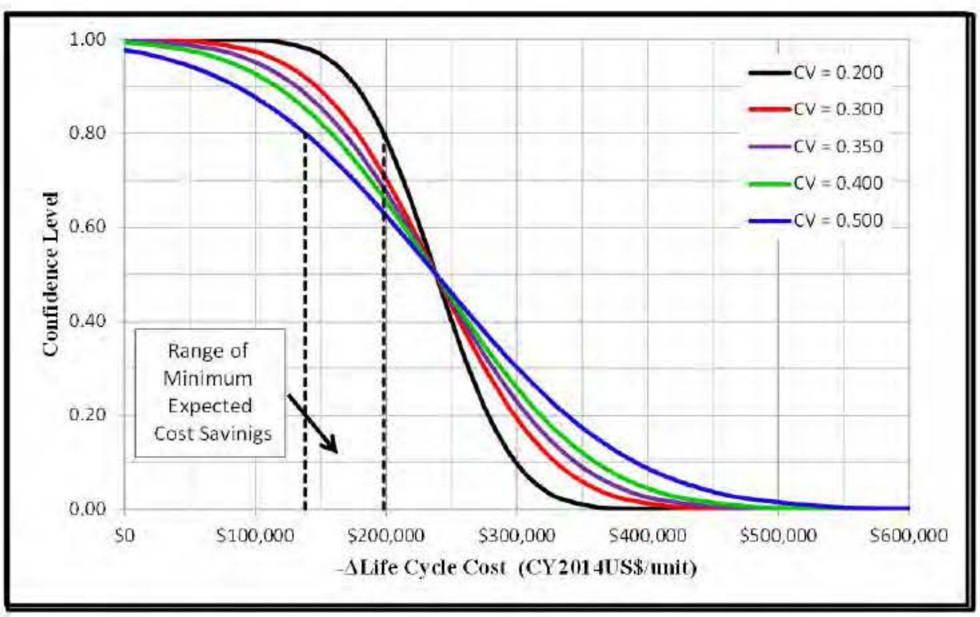

Figure 17. Diesel EDH ΔLCC Probability Distribution

The range of minimum expected differential savings with respect to the baseline is between $138,143 and $198,164 based on a present value point estimate of $238,178. The point estimate minus 20% yields a value of $190,542 which is toward the less conservative end of the range. The additional program risk makes this option less attractive when compared to the Diesel DDH architecture making the potential benefits of readiness due to equipment familiarity a more expensive proposition; yet this option still shows significant savings over the baseline. The point of present value equivalence occurs between confidence levels of 97.725% and 100.0% for the range of coefficients of variance considered. Costs are in CY2014US$ per unit.

4. Analysis Summary

Summary of the cost variation is shown in Figure 18.

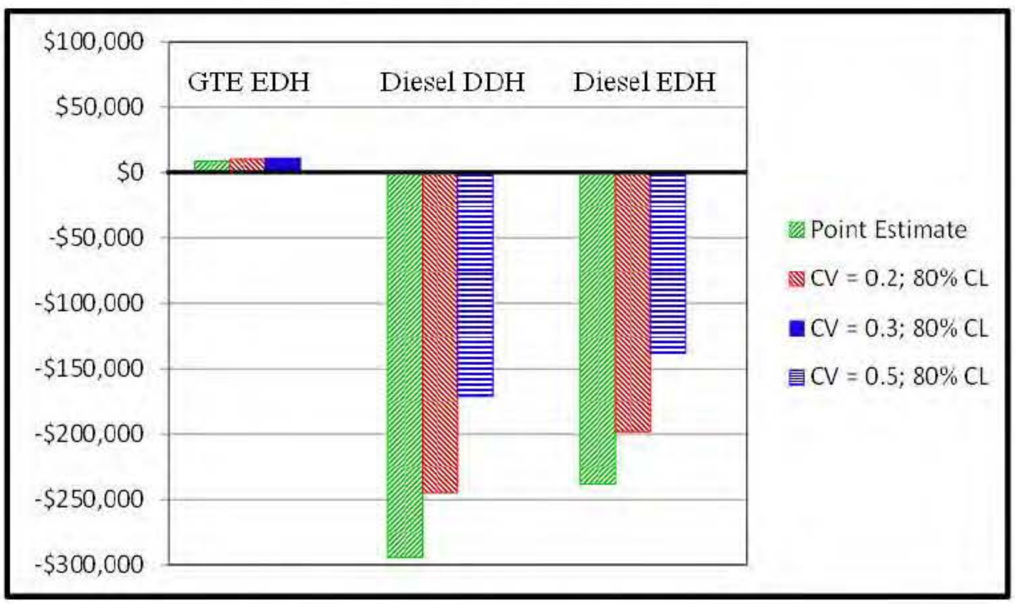

Figure 18. Summary of Cost Variation

The eSBM analysis aids in bounding the projected costs of a program. In this case the variation in cost generated by the analysis does not create ambiguity in the rank order of alternatives based on the cost point estimates. Between the two diesel architectures, the cost difference is about one standard deviation with respect to the direct hydraulic drive design.

D. RECOMMENDATION

The architecture recommendation is based on the systems cost and the ability of the system to meet requirements. Requirements satisfaction will compose 65% of the weighting criteria, cost 25%, and reliability 10%. Requirements satisfaction is further delineated with performance requirements 33%, mass 15%, and envelope 15%, and sound pressure levels at the operators' panel 2%. The gas turbine architectures using the baseline power plant meet all the requirements except the peak power requirement. Diesel power plants that meet the peak power requirement are not viable from a mass perspective so the power plants considered are comparable with respect to requirements compliance. In addition, the diesel system pneumatics are not as robust or flexible as those of the GTE systems and may require the use of kits to attain full functionality. Though projected to mission objectives, the diesel systems pneumatic capability is less than that of the gas turbine systems—this deficiency is reflected by a 10-point differential in the raw performance requirement score. The mass and envelope raw scores are based on the predicted percentage of the range between the threshold and maximum requirement. The objective level for volume is established using the baseline packing density in conjunction with the objective mass.

Cost data raw scores are the ratio of the point estimate and the 80% confidence level range of differential life cycle costs for all architectures expressed as a percentage. Reliability raw score is calculated as 100 minus the reliability expressed in percentage. Raw scores for SPL at the operator panel are determined as 100 minus the percentage of an 8-hour shift an operator can work at the panel given the sound pressure level.

A bonus of 5% of the total weighted score of the baseline is awarded to systems that can be utilized in the hangar and provide an extra benefit to readiness. Table 13 presents scoring with cost structure presented in the previous section.

The scoring shows that with only a 25% weighting, cost is still the primary factor. Interestingly the bonus points allotted the electric hydraulic design is enough to overcome the cost increase associated with the configuration change in the gas turbine systems.

Unlike the gas turbine systems, a premium still is required for the electric drive hydraulics on the diesel architectures.

Table 13. System Concept Scoring

Criteria	Weight	Baseline		GTE EDH		Diesel DDH		Diesel EDH	
		raw	weighted	raw	weighted	raw	weighted	raw	weighted
Performance Requirements	33%	10.0	3.3	10.0	3.3	20.0	6.6	20.0	6.6
Mass	15%	37.1	5.6	45.0	6.8	91.9	13.8	94.7	14.2
Volume	15%	18.0	2.7	18.0	2.7	48.2	7.2	48.2	7.2
ΔLCC	25%	97.0	24.3	97.6	24.4	14.0	3.5	29.8	7.5
Reliability	10%	10.0	1.0	10.0	1.0	4.0	0.4	4.0	0.4
SPL	2%	50.0	1.0	50.0	1.0	0.0	0.0	0.0	0.0
Hangar Use	Bonus	0.0	0.0	-1.9	-1.9	0.0	0.0	-1.9	-1.9
SCORE	↓ Better		37.8		37.2		31.5		34.0

As shown in Table 14, the point of equivalence between the systems requires a bonus of about 12% of the weighted baseline score. The diesel power plant with direct drive hydraulics remains the dominant system.

The recommended system essentially replaces the gas turbine engine with a diesel power plant. If the total mass and pneumatic power challenges cannot be mitigated the best option becomes the gas turbine engine with electric drive hydraulics.

Table 14. Adjusted System Concept Scoring

Criteria	Weight	Baseline		GTE EDH		Diesel DDH		Diesel EDH	
		raw	weighted	raw	weighted	raw	weighted	raw	weighted
Performance Requirements	33%	10.0	3.3	10.0	3.3	20.0	6.6	20.0	6.6
Mass	15%	37.1	5.6	45.0	6.8	91.9	13.8	94.7	14.2
Volume	15%	18.0	2.7	18.0	2.7	48.2	7.2	48.2	7.2
ΔLCC	25%	97.0	24.3	97.6	24.4	14.0	3.5	29.8	7.5
Reliability	10%	10.0	1.0	10.0	1.0	4.0	0.4	4.0	0.4
SPL	2%	50.0	1.0	50.0	1.0	0.0	0.0	0.0	0.0
Hangar Use	Bonus	0.0	0.0	-4.4	-4.4	0.0	0.0	-4.4	-4.4
SCORE	↓ Better		37.8		34.8		31.5		31.5

E. CHAPTER SUMMARY

The influence of specific fuel consumption on the differential life cycle cost is the prevailing factor in concept scoring. Using large coefficients of variance for the application with a cost weighting factor of 25% does not influence the diesel direct drive architecture rank established by the point estimate. This architecture is dominant and will be significantly less expensive than the baseline at a confidence level of 80%.

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VI. CONCLUSIONS

A. KEY POINTS

Net present value analysis of procurement and operation and support costs revealed the power plant as the component that offers the greatest opportunity for cost reduction. Diesel power plants' low specific fuel consumption relative to the baseline gas turbine engine proved to be the predominant factor followed by power plant procurement costs. These savings cannot be realized without risk mitigation with respect to system mass of the diesel architectures. Also the most favorable architecture based on the selection criteria does not allow in-hangar operations, a feature that could potentially aid system maintenance during peacetime operations. The diesel engine architectures also require greater interaction with the motor pool and would create a more complex logistics environment. Nonetheless, the potential life cycle cost savings of $170,000 (CY2014) per unit is substantial and warrants serious consideration against the risks and logistics challenges.

If the challenges of the diesel architectures preclude their use then the next best value is the baseline with electric drive COTS hydraulic pumps that allow in hangar operation. Any future GTE architectures should examine means to increase fuel efficiency. Specific fuel consumption of the baseline system is simply not acceptable. The GTE systems have a greater system mass trade space to explore higher compression ratio engines as well as intercooling and regeneration. The point estimate cost evaluation shows that the GTE system does not have to match the fuel economy of a diesel but does need to improve significantly from current levels. Moving from an aeroderivative engine to an industrial engine reduces the fixed capital expense of the engine. Trade studies would have to determine the proper balance of fuel economy and initial cost within mass trade space. Future gas turbine engines also need to be compatible with engine wash systems being utilized by the aircraft for cleaning the compressor and heat exchangers if any are employed.

B. AREAS OF FURTHER RESEARCH

Understanding of the requirements is crucial when developing products. Potential exists for over-design by multiple applications of safety factors and design margin from the aircraft systems to the support equipment. Study of the aircraft maintenance practices and the actual demands on the support equipment may allow for reductions in power plant capacity and thus less fuel consumption. A specific area of study is the pneumatic power requirement to air start the T-700 engine. Only the maximum flow for the starter is provided in the specifications, understanding the minimum flow may allow for mass reduction in diesel architectures.

If gas turbine engines are to be used for ground power units in the future low labor intensive or automated techniques for compressor and heat exchanger cleaning need to be developed. This is essential to increasing and maintaining better gas turbine engine fuel efficiency.

APPENDIX.
DIFFERENTIAL LIFE CYCLE COST NET PRESENT VALUE

Table 15. GTE EDH ΔLCC Net Present Value (from Holland, Watson, and Wilkinson 1984; Newnan 1983)

Program Profile			Base Year 2014	Differential Costs from Baseline (Thousands of US$CY2014)										
Units Procured	Units in Service	Units with Overhaul Burden	Period (yrs)	Fixed Capital	Parts	Overhaul	Fuel	Waste	Total O&S	Total Δ Cost	Discount Rate	Discount Factor	Discount Value	Net Present Value ($CY2014)
144	0	0	0	(6804)	0	-	-	-	0	(6804)	1.00	1.0000	(6804)	($6,804,000)
144	144	0	1	(6804)	(368)	-	-	-	(368)	(7172)	1.00	0.9901	(7101)	($13,904,911)
144	288	28	2	(6804)	(736)	-	-	-	(736)	(7540)	1.00	0.9803	(7391)	($21,296,186)
144	432	86	3	(6804)	(1104)	-	-	-	(1104)	(7908)	1.00	0.9706	(7675)	($28,971,380)
144	576	173	4	(6804)	(1472)	-	-	-	(1472)	(8276)	1.90	0.9275	(7676)	($36,646,885)
0	720	288	5	0	(1840)	-	-	-	(1840)	(1840)	1.90	0.9102	(1674)	($38,321,259)
0	720	432	6	0	(1840)	-	-	-	(1840)	(1840)	2.50	0.8623	(1586)	($39,907,540)
0	720	547	7	0	(1840)	-	-	-	(1840)	(1840)	2.50	0.8413	(1548)	($41,455,132)
0	720	634	8	0	(1840)	-	-	-	(1840)	(1840)	3.00	0.7894	(1452)	($42,907,329)
0	720	691	9	0	(1840)	-	-	-	(1840)	(1840)	3.00	0.7664	(1410)	($44,317,229)
0	720	720	10	0	(1840)	-	-	-	(1840)	(1840)	3.00	0.7441	(1369)	($45,686,065)
0	720	720	11	0	(1840)	-	-	-	(1840)	(1840)	3.60	0.6777	(1247)	($46,932,776)
0	720	720	12	0	(1840)	-	-	-	(1840)	(1840)	3.60	0.6542	(1203)	($48,136,166)
0	720	720	13	0	(1840)	-	-	-	(1840)	(1840)	3.60	0.6314	(1162)	($49,297,739)
0	720	720	14	0	(1840)	-	-	-	(1840)	(1840)	3.60	0.6095	(1121)	($50,418,949)
0	720	720	15	0	(1840)	-	-	-	(1840)	(1840)	3.60	0.5883	(1082)	($51,501,198)
0	720	720	16	0	(1840)	-	-	-	(1840)	(1840)	3.60	0.5679	(1045)	($52,545,839)
0	720	720	17	0	(1840)	-	-	-	(1840)	(1840)	3.60	0.5481	(1008)	($53,554,181)
0	720	720	18	0	(1840)	-	-	-	(1840)	(1840)	3.60	0.5291	(973)	($54,527,483)
0	720	720	19	0	(1840)	-	-	-	(1840)	(1840)	3.60	0.5107	(939)	($55,466,964)
0	720	720	20	0	(1840)	-	-	-	(1840)	(1840)	3.60	0.4930	(907)	($56,373,799)
0	576	547	21	0	(1472)	-	-	-	(1472)	(1472)	3.60	0.4758	(700)	($57,074,058)
0	432	288	22	0	(1104)	-	-	-	(1104)	(1104)	3.60	0.4593	(507)	($57,581,002)
0	288	86	23	0	(736)	-	-	-	(736)	(736)	3.60	0.4433	(326)	($57,907,221)
0	144	28	24	0	(368)	-	-	-	(368)	(368)	3.60	0.4279	(157)	($58,064,662)
0	28	0	25	0	(72)	-	-	-	(72)	(72)	3.60	0.4131	(30)	($58,094,212)

Table 16. Diesel DDH ΔLCC Net Present Value (from Holland, Watson, and Wilkinson 1984; Newnan 1983)

Program Profile			Base Year 2014	Differential Costs from Baseline (Thousands of US$CY2014)										Net Present Value ($CY2014)
Units Procured	Units in Service	Units with Overhaul Burden	Period (yrs)	Fixed Capital	Parts	Overhaul	Fuel	Waste	Total O&S	Total Δ Cost	Discount Rate	Discount Factor	Discount Value	
144	0	0	0	10224	0	0	0	0	0	10224	1.00	1.0000	10224	$10,224,000
144	144	0	1	10224	(306)	0	2876	(12)	2558	12782	1.00	0.9901	12655	$22,879,461
144	288	28	2	10224	(612)	(43)	5752	(24)	5074	15298	1.00	0.9803	14996	$37,875,568
144	432	86	3	10224	(918)	(131)	8628	(36)	7544	17768	1.00	0.9706	17245	$55,120,528
144	576	173	4	10224	(1224)	(263)	11504	(48)	9969	20193	1.90	0.9275	18729	$73,849,494
0	720	288	5	0	(1530)	(437)	14380	(60)	12353	12353	1.90	0.9102	11243	$85,092,899
0	720	432	6	0	(1530)	(656)	14380	(60)	12134	12134	2.50	0.8623	10463	$95,556,271
0	720	547	7	0	(1530)	(830)	14380	(60)	11960	11960	2.50	0.8413	10061	$105,617,580
0	720	634	8	0	(1530)	(962)	14380	(60)	11828	11828	3.00	0.7894	9337	$114,954,450
0	720	691	9	0	(1530)	(1049)	14380	(60)	11741	11741	3.00	0.7664	8999	$123,953,058
0	720	720	10	0	(1530)	(1093)	14380	(60)	11697	11697	3.00	0.7441	8704	$132,656,813
0	720	720	11	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.6777	7927	$140,584,047
0	720	720	12	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.6542	7652	$148,235,816
0	720	720	13	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.6314	7386	$155,621,694
0	720	720	14	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.6095	7129	$162,750,920
0	720	720	15	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.5883	6881	$169,632,412
0	720	720	16	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.5679	6642	$176,274,779
0	720	720	17	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.5481	6412	$182,686,330
0	720	720	18	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.5291	6189	$188,875,086
0	720	720	19	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.5107	5974	$194,848,788
0	720	720	20	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.4930	5766	$200,614,910
0	576	547	21	0	(1224)	(830)	11504	(48)	9402	9402	3.60	0.4758	4474	$205,088,461
0	432	288	22	0	(918)	(437)	8628	(36)	7237	7237	3.60	0.4593	3324	$208,412,268
0	288	86	23	0	(612)	(131)	5752	(24)	4985	4985	3.60	0.4433	2210	$210,622,475
0	144	28	24	0	(306)	(43)	2876	(12)	2516	2516	3.60	0.4279	1076	$211,698,921
0	28	0	25	0	(60)	0	559	(2)	497	497	3.60	0.4131	205	$211,904,370

Table 17. Diesel EDH ΔLCC Net Present Value (from Holland, Watson, and Wilkinson 1984; Newnan 1983)

Program Profile			Base Year 2014	Differential Costs from Baseline (Thousands of US$CY2014)										Net Present Value ($CY2014)
Units Procured	Units in Service	Units with Overhaul Burden	Period (yrs)	Fixed Capital	Parts	Overhaul	Fuel	Waste	Total O&S	Total Δ Cost	Discount Rate	Discount Factor	Discount Value	
144	0	0	0	1922	0	0	0	0	0	1922	1.00	1.0000	1922	$1,922,400
144	144	0	1	1922	(306)	0	2876	(12)	2558	4480	1.00	0.9901	4436	$6,358,455
144	288	28	2	1922	(612)	(43)	5752	(24)	5074	6996	1.00	0.9803	6858	$13,216,536
144	432	86	3	1922	(918)	(131)	8628	(36)	7544	9466	1.00	0.9706	9188	$22,404,045
144	576	173	4	1922	(1224)	(263)	11504	(48)	9969	11892	1.90	0.9275	11029	$33,433,466
0	720	288	5	0	(1530)	(437)	14380	(60)	12353	12353	1.90	0.9102	11243	$44,676,871
0	720	432	6	0	(1530)	(656)	14380	(60)	12134	12134	2.50	0.8623	10463	$55,140,243
0	720	547	7	0	(1530)	(830)	14380	(60)	11960	11960	2.50	0.8413	10061	$65,201,552
0	720	634	8	0	(1530)	(962)	14380	(60)	11828	11828	3.00	0.7894	9337	$74,538,422
0	720	691	9	0	(1530)	(1049)	14380	(60)	11741	11741	3.00	0.7664	8999	$83,537,030
0	720	720	10	0	(1530)	(1093)	14380	(60)	11697	11697	3.00	0.7441	8704	$92,240,785
0	720	720	11	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.6777	7927	$100,168,019
0	720	720	12	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.6542	7652	$107,819,788
0	720	720	13	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.6314	7386	$115,205,666
0	720	720	14	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.6095	7129	$122,334,892
0	720	720	15	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.5883	6881	$129,216,384
0	720	720	16	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.5679	6642	$135,858,751
0	720	720	17	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.5481	6412	$142,270,302
0	720	720	18	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.5291	6189	$148,459,058
0	720	720	19	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.5107	5974	$154,432,760
0	720	720	20	0	(1530)	(1093)	14380	(60)	11697	11697	3.60	0.4930	5766	$160,198,882
0	576	547	21	0	(1224)	(830)	11504	(48)	9402	9402	3.60	0.4758	4474	$164,672,433
0	432	288	22	0	(918)	(437)	8628	(36)	7237	7237	3.60	0.4593	3324	$167,996,240
0	288	86	23	0	(612)	(131)	5752	(24)	4985	4985	3.60	0.4433	2210	$170,206,447
0	144	28	24	0	(306)	(43)	2876	(12)	2516	2516	3.60	0.4279	1076	$171,282,893
0	28	0	25	0	(60)	0	559	(2)	497	497	3.60	0.4131	205	$171,488,342

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